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Transportation's Research Needs

However remarkable have been the strides of engineering progress in the past, the opportunities before us command all the experience, knowledge, breadth of vision, and foresight that this body represents. Many problems of practical importance have already become the subject of research projects under the Society's auspices. Railroad transportation alone invites your best ingenuity. We have only scratched the surface of improvements, economies, and efficiencies which almost every department of railroad transportation will require in the future.

W. W. ATTERBURY

(President, Pennsylvania Railroad System, in an address at the Altoona Regional Meeting of the A.S.M.E., October 5–7, 1925)

NOVEMBER 1925

THE MONTHLY JOURNAL PUBLISHED BY THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Setting exact standards for the oxy-acetylene process

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CONTENTS OF THIS ISSUE

American Transportation ... *Samuel Rea* ... 871
The Rustproofing of Materials ... *M. E. McDonnell* ... 875
The Locomotive Testing Plant and Its Influence on Steam-Locomotive Design ... *L. H. Fry* ... 881
Lubricants and Lubrication ... *James Duguid* ... 887
Future of the Machine-Tool Industry ... *O. B. Iles* ... 894
Correction to Boyle's Law for High-Pressure Gas ... *D. E. Silcox* ... 897
A System of Limits for Different Kinds of Fits ... *C. D. Albert* ... 901
Measuring Systems and Tolerance Limits ... *P. J. Darlington* ... 903
The Prime Cause of Friction ... *E. W. Davis* ... 906
An Instrument for Recording Vibrations ... *L. H. Young* ... 907
Boiler-Feedwater Purification ... *A. S. Behrman* ... 909
The Savenius Wing Rotor ... *Alexander Klemin* ... 911
Logging Machinery Used on the Pacific Coast ... *W. C. Shaw* ... 913
Steam-Consumption Tests on a Back-Pressure Turbine ... *A. Stodola* ... 915
American Society for Steel Treating (Résumé of meeting papers) ... 916
Test Code on Instruments and Apparatus ... 920
Centerless Grinding ... 943
Machine-Shop Sessions at New Haven ... 946
Distinctive Railroad Meeting at Altoona ... 951

DEPARTMENTAL

Survey of Engineering Progress ... 927
A Review of Attainment in Mechanical Engineering and Related Fields
Engineering and Industrial Standardization ... 939
Correspondence ... 941
Critical Bearing Pressures; Concerning Supersaturation
Editorial Notes ... 948
Gold from Mercury; Novelty of Standardization; "Railroad Men;" Patent Office Procedure; Pulitzer Race of 1925; Cleveland Safety Congress
Library Notes and Book Reviews ... 954
The Engineering Index ... 123-EI ... Late Items ... 958

ADVERTISING

Display Advertisements ... 1
Professional Engineering Service Section ... 116
Alphabetical List of Advertisers ... 144
Opportunity Advertisements ... 121
Classified List of Mechanical Equipment ... 124

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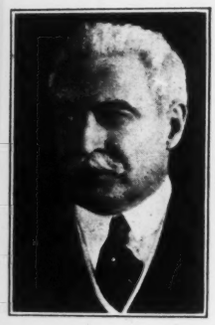
L. H. Fry

O. B. Iles



Samuel Rea

C. D. Albert

L. H. Young

Contributors to this Issue

Samuel Rea who has just retired from the presidency of the Pennsylvania Railroad System, has held that office since 1912. At the age of fifteen Mr. Rea left school to become a chainman on the Pennsylvania lines. From 1879 to 1889 he was engineering assistant to the vice-presidents of the Pennsylvania System, leaving to become vice-president and engineer of the Maryland Central during the construction of the Baltimore Belt Line. He returned to the Pennsylvania in 1892. Mr. Rea had full charge of the great project which brought the Pennsylvania lines into the heart of New York City and of the associated Hell Gate Bridge development.

* * * * *

M. E. McDonnell is chief chemist of the Pennsylvania Railroad System. He was graduated from Pennsylvania State College in 1893 with the degree of B.S. He received his M.S. in 1896 and his Ph.D. from the University of Kiel, Germany. He entered the service of the Pennsylvania Railroad System in 1899 as assistant chemist, and in 1907 was appointed bacteriologist; in 1916, assistant chief chemist; and in 1921, chief chemist.

* * * * *

L. H. Fry is the metallurgical engineer of the Standard Steel Works Co. In 1897 Mr. Fry began work in the erecting shop of the Baldwin Locomotive Works, later becoming sales engineer and engineer of tests. He was technical representative of the works in London from 1905 to 1913, and since then has been connected with the company in this country as superintendent of product and in his present position as metallurgical engineer.

* * * * *

James Duguid, engineer of tests with the Galena-Signal Oil Co., Franklin, Pa., since 1918, served his apprenticeship as a machinist in Canada. For three years he was a special student in the technical school of Toronto. He served as general foreman in the machine shops of the Grand Trunk at Stratford and later at Toronto, Canada. He then became master mechanic of the Eastern Lines of the same railroad, resigning to take the position of assistant mechanical superintendent of the Central Vermont Railroad.

O. B. Iles, president of the International Machine Tool Co., Indianapolis, Ind., was graduated from DePauw University in 1894. He was admitted to the bar in Indiana in 1896. From 1899 to 1903 he was connected with the Arthur Jordan Co., Indianapolis, and from 1903 to 1906 was manager of the Capital Gas Engine Co., also in that city. At that time he became connected with the International Machine Tool Co. of which firm he is now president. Mr. Iles is president of the National Machine Tool Builders' Association.

* * * * *

Calvin D. Albert is professor in charge of the department of machine design, Sibley School of Mechanical Engineering, Cornell University. He received his M.E. from Cornell in 1902 and in 1904 joined the teaching staff of Sibley College. From 1917 to 1919 Professor Albert was associated with the United States Shipping Board, Emergency Fleet Corporation, then returning to Cornell.

* * * * *

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* * * * *

A. S. Behrman is a 1914 graduate of the University of Kentucky. He spent several years in the Philippines in water survey investigations of the Bureau of Science. During the war he was officer in charge of field purification of water in the zone of advance of the First Army. Since the war he has been chief chemist of the International Filter Co., Chicago.

* * * * *

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* * * * *

L. H. Young was graduated from Massachusetts Institute of Technology in 1915 and then served for two years in the physics department of the Institute. From 1917 to 1919 he was in the aviation service of the U. S. Navy and since that time has been a member of the M.I.T. physics department staff.

* * * * *

Alexander Klemin is associate professor of aeronautics in New York University. From 1915 to 1917 he had charge of the department of aeronautics at M.I.T., and, during the war, of the aeronautic research section of McCook Field.

* * * * *

W. C. Shaw was until recently chief engineer of the Smith & Watson Iron Works, Portland, Ore., and since 1900 has been associated with various firms on the Pacific Coast.

* * * * *

D. E. Silcox is in the producing department of the Standard Oil Co. of California, Whittier, Cal.

A.S.M.E. Annual Meeting, November 30–December 4, 1925, New York, N. Y.

About forty-five papers will be presented at the coming Annual Meeting, and in accordance with the procedure followed last year a special issue of MECHANICAL ENGINEERING will be put in the mails on November 15 containing a portion of the papers and synopses of the remainder. Honorary Membership will be conferred upon Secretary of Commerce Hoover and upon Worcester R. Warner, Past-President of the A.S.M.E. As usual, the National Exposition of Power and Mechanical Engineering will be held in the Grand Central Palace parallel to the Meeting.

The current issues of the A.S.M.E. News are carrying the details of the Meeting.

MECHANICAL ENGINEERING

Volume 47 November, 1925 No. 11

American Transportation

The Growth of American Railroads and the Part They Are Playing in the Progress of the Nation

By SAMUEL REA,[1] PHILADELPHIA, PA.

I DEEM IT an especial honor to speak before this distinguished body upon the development of the transportation industry of our country. The growth of American railroads, the unrivaled perfection which they have reached, both in standards of construction and performance, and the part which they have played and are playing in the progress of the nation, rank among the most important achievements of the engineering professions—achievements, moreover, to which every branch, civil, mechanical, electrical, and chemical, has contributed its share.

In equal degree the history of our railroad evolution and growth stands as a preëminent tribute to the foresight, courage, and enterprise of American business men, financiers, and the investing public, because their functions in the creation of our matchless rail highways have been no less vital and fundamental than have those of science and the technical professions.

The hundredth anniversary of the establishment of transportation by steam railroads is being celebrated this year in England. The date usually fixed for the initial operation in this country is 1827, so that the American railroad industry is now 98 years old. My own personal experience in the railroad and transportation field dates from 1871, and covers a period of 54 years. It has therefore been my privilege and opportunity to have observed, in active service, the progress and advance of transportation during more than half of the entire world history of railroad development, and that period, of course, covers far more than half of the evolution of steam railroads into the highly perfected instruments of public service which we know today.

I may add, as perhaps an interesting reflection upon how fast the world moves and how much can be compressed into the experiences of a single lifetime, that I completed a quarter of a century of railroad work before motor cars were ever heard of by any one except a few experimenters who called them "horseless carriages," and over one-third of a century before flying by mechanical power was anything more than a dream which most people thought impossible of realization.

After the progress I have seen with my own eyes, merely in the transportation field alone, I would consider any man extremely rash who would attempt to set a limit to the possible further achievements of the engineering professions and of science in general toward the continued advancement and betterment of humanity.

The subject which has been assigned to me, American Transportation, covers so vast a scope that in a brief talk it is possible to do little more than merely touch upon the most outstanding facts and indicate a few of the principal milestones in the road of progress.

Railroad Development in the United States

While it is true that railroads had their start in England, they have had by far their greatest and most important development here in the United States. Indeed, the story of American railroads has been written large and boldly upon practically every page of our company's history; while as factors in the economic life and advancement of the people, and in our national progress, the importance of the part which they have played has not even been approached by the railroads of any other nation.

The total railway plant of the world embraces at the present time something over 600,000 miles of line. Of these, some 260,000 miles, or not far from 40 per cent, are within the boundaries of the Continental United States. As we have only 5 to 6 per cent of the world's land area and population, it is therefore evident that we have some six or eight times as much railroad plant per capita here in America as the world's average.

Our use of railroad facilities per capita is on even a larger comparative scale. The average American citizen, for example, utilizes from four to five thousand ton-miles of railroad service per year. That is to say, the railroads of America transport a ton of freight that many miles for each man, woman, and child in the country. Compared with this, the average citizen of Great Britain uses 300 ton-miles or less; of France, perhaps 400 ton-miles; while the prewar German consumed some 600 ton-miles of rail service annually. This means that the per capita use of railroad service in America, measured in ton-miles of freight service, is from seven to fifteen times as great as in the most highly developed countries of Europe, and if we took the world average, including the less developed countries, the comparison would naturally be still more striking.

The difference, of course, is explainable largely by the relatively great distances in this country between the sources of raw material, the centers of production and the centers of consumption, coupled with our highly developed system of specialized mass industry. It measures the extremely high degree of our dependence upon railroad transportation, not only for the state of national progress which we have already reached, but for its day-to-day maintenance and for the continuance of our progress in the future.

Mechanical transportation, as afforded by the railroads, opened up the whole vast territory embraced within our national boundaries to settlement and development, and did so with a rapidity never known before in human history. It continues to be the principal physical means by which the economic life of the entire nation is sustained and the continuity of its commerce, industry and agriculture, and every form of social activity insured.

No one can be more willing than myself to recognize the advent of other forms of transportation or to appraise liberally their value and importance. I am, however, profoundly convinced that their eventual effect will be greatly to increase, and in no sense diminish the vital importance and necessity of the railroads. Motor transportation, it is quite true, has already required extensive changes in railroad methods and practices, and in conjunction with air transport may not improbably bring about still further changes of important character. It is, however, my firm belief that, as the basic carriers of the nation, the railroads will indefinitely retain the position of fundamental supremacy and importance.

The Sound Financial Basis of American Railroads

The railroads of the United States as they stand today represent on their books an investment in physical property—that is, road and equipment—of some twenty-two billions of dollars. The progress of physical valuation by the Interstate Commerce Commission, thus far, gives every warrant for expectation that these book figures of original cost will be much more than sustained. It is very questionable whether these great properties could be produced anew today, under existing conditions, for double the sums at which they are carried upon the balance sheets of their respective corporations. The real value of the railroads to the country as a whole is totally incapable of expression in dollars at all, for the reason which I have just stated, viz., that they are an absolutely indispensable

[1] Retired President, Pennsylvania Railroad Company.

Address delivered at the Altoona Regional Meeting of The American Society of Mechanical Engineers, Altoona, Pa., October 5–7, 1925.

instrument for the carrying on of the daily life and commerce of our people.

The capitalization of our railroads, an entirely distinct thing from cost or physical valuation, is much less than the investment figures. The total amount of railroad stocks and bonds at present outstanding in the hands of the public is only some seventeen billions of dollars. The difference between that sum and the figure of twenty-two billion dollars for investment in physical property invites attention to the vast degree of undercapitalization, the very reverse of "water," which is characteristic of the financial structure of American railroads as a whole. This difference or "spread" will be increased rather than diminished when the federal valuation work is completed.

It is safe and conservative to say that there is no other great form of industry or enterprise in the world which has been built upon a sounder financial basis, or which possesses greater fundamental strength and integrity in financial structure than those that characterize our railroads. Should the solvency of these vast properties as a whole ever be threatened, the cause will not lie in any flaws or weaknesses inherent in themselves, but rather in elements beyond the control of their managements, such, for example, as a long-continued abuse of the governmental power to regulate rates, and encroachments on the duties and responsibilities of directors. Such a disaster, however, I hasten to say, has in my opinion no longer the slightest likelihood of occurring. I am glad to voice my belief that we are past the principal crisis with respect to the difficulties and errors of governmental regulation and are definitely on the way to a condition in which constructive and helpful policies will distinctly dominate the regulative field.

Present Trend of Governmental Regulation

A steady improvement in the spirit and purpose of regulation has been quite evident in recent years, and more particularly since the termination of the temporary wartime control of the railroads. It can reflect but one thing: that is, the more friendly attitude of public opinion arising through the widespread better understanding of railroad problems, and of the vitally essential character of railroad service, which has come about so largely in the last decade.

Personally, I have never believed at any time that railroad baiting was really popular in the sense that it represented the will or wishes of a majority of our people. For a long period, however, the friends of the railroads—those who understood and appreciated their value and necessity—were inarticulate. Only their professional enemies, those who thrived and made political capital by attacking "big business" and success, made themselves heard. Happily, all that is now profoundly changed. The public has been thoroughly aroused to the dangers of harsh and repressive regulation of the railroads, and has manifested unmistakable impatience and resentment at any tendency toward continued abuse and injury of these great public servants, for the purpose of furthering political ends or personal ambitions.

Higher Returns Justified

But, gratifying as is the change we are witnessing in these respects, it will never do to forget that railroad regulation is still far from perfect, and that we have a long way to go before it can be regarded as upon a truly sound basis. We cannot shut our eyes, for instance, to the fact that the provisions of the Transportation Act of five years ago, requiring the establishment of rates which will yield a fair and reasonable return upon the value of the railroads as a whole, have never yet been fully carried out. Not even the inadequate return of 5¾ per cent, tentatively set by the Interstate Commerce Commission, has yet been realized.

In my opinion two things are required in this regard. One is to raise the contemplated rate of return to a materially higher figure than has yet been set. Does not that certainly seem justified in view of the fact that returns of 7 and 8 per cent on such utilities as gas, electric light, and traction properties have repeatedly been declared reasonable, and that such institutions as national banks, which were under governmental regulation and supervision long before the railroads, are permitted and encouraged to earn returns far higher still?

The second requisite, after having raised the proposed return to an adequate level, is to instill into our regulative bodies the confidence, courage, and initiative necessary to make such return a reality. When, and if, that is done—and it will have to come gradually—railroad management will be unfettered, and will be again in a position to exercise its true functions properly, and what we call the "railroad problem" will cease to trouble the American people.

The question all harks back to further education and enlightenment of public opinion, and to increased insistence by the business and professional circles of the country, upon recognition of the fact that our national prosperity in the long run can only rest fundamentally and soundly upon the basis of progressive and prosperous transportation.

Developments in Railroading Since the A.S.M.E. Altoona Meeting in 1883

General Atterbury and Mr. Lee have already mentioned the fact that this Society held its Third Annual Meeting here at Altoona in 1883, or 42 years ago. In joining them to extend to you a welcome back to this great seat of the mechanical engineering profession, as applied to railroad work, it occurred to me that possibly you would be interested in a very brief reference to some of the general trends in railroading which have taken place during that interval.

Forty-two years ago the pioneering era, at least in the East, was nearing its end and our railroads were considerably more than one-third built, in so far as length of line mileage is concerned. The progress which has taken place since that time has largely consisted in steadily continued improvement of the plant, such as additional line and tracks and enlarging old and building new yards, and in the more intensive and concentrated use of facilities. This, of course, has been particularly true in the last ten or fifteen years, during which the construction of new line has been only very slight, while the additions to the previously existing plant have been enormous.

Summarizing the developments which have occurred since your last meeting in Altoona, I may point out that in the forty-two years line mileage has increased about 150 per cent and the capital has about tripled, while the use of facilities, or volume of railroad service rendered, measured in ton-miles and passenger-miles, has been approximately multiplied by ten. The freight train-miles have less than doubled, but the tons per train have been multiplied by four, and the length of the haul for each ton of originating freight considerably increased.

These comparisons, in necessarily very round figures, outline the story of railroad development in the last four decades. They reflect, of course, the achievements with which we are all familiar—far more powerful locomotives, larger and stronger cars, and longer trains, together with heavier rails, improved roadbed, strengthened bridges, larger yards and improved structures generally, necessary to handle mass transportation on a constantly growing scale. In all of these achievements the railroads of our country have led and are leading the world, which accounts for the fact that we have the cheapest rates and the best service, and at the same time pay the highest wages.

Factors Controlling the Future of American Transportation

Looking to the future of American transportation, we have several controlling factors to consider, and, if we are to form accurate conclusions, must properly interpret their reaction and interaction upon one another.

Line Mileage. The first of these factors, in my judgment, is that the present mileage of our railroads covers pretty thoroughly those portions of the country most desirable and readily adapted to economic exploitation, i.e., the portions naturally suited to agricultural, industrial, or mineral development, and to the support of large centers of population. It is true that there still are great areas within the national boundaries not traversed by rail lines. Examination, however, will show that these consist almost entirely, if not altogether, of the very rugged mountain regions or desert and semi-desert sections. Practically all of the naturally arable land of the United States, and a very large proportion of that capable of development by irrigation, is now accessible by rail. That is true also of our timber lands and of most, if not all, of our valuable mineral deposits, including coal and oil. Indeed, as we are well aware and should not hesitate to acknowledge, some of the latter are

doubtless developed at the present time beyond needs immediately in sight.

We therefore have a picture of a great nation already liberally supplied with lines of railroad communication readily adapted to such further development as the progress and growth of the country may require. A glance at any railroad map will confirm this viewpoint.

Motor-Vehicle Transport. The second factor broadly bearing on the transportation future is the motor vehicle. As a railroad man I have not the slightest fear in admitting that for many purposes motor cars are capable of furnishing short-distance transportation to better advantage, and with greater economy and efficiency, than it can be done by rail. The country as a whole, however, I think, is coming to the realization that the motor car's true function, especially as a commercial carrier, is chiefly as a feeder of the railroads and as a connecting link between the existing rail lines. Save in rare instances, it is, and must indefinitely remain, a far less efficient, less satisfactory, and less economical instrument than the railroads for long-distance or bulk transportation of either passengers or freight. Incidentally, I wish to express the further view that, in the long run, motor cars, through the vastly increased business activity which they have brought about in so many directions, will create for the railroads much more new traffic than they will ever by any possibility take away through direct competition. Indeed, they have been doing this ever since they became an important influence in the manufacturing and transportation fields. The real problem as between railroads and motor cars is not one of competition but of coördination, to the end that each of these agencies of transportation may be free to develop and progress in its proper field, and in that manner best serve the public needs.

Air Transport. The third factor for consideration is air transport, still largely an unknown quantity. If I were to hazard a guess, however, it would be that its commercial development as an agency of passenger service will, for at least a long time in the future, be confined to very high-grade de luxe transportation for people willing to pay necessarily high rates in return for exceptional speed, the saving in time, and the novelty and distinction of a mode of travel open only to the few.

As a carrier of goods, air transport is quite likely to develop, on a considerable scale, in the field of high-class mail and express, and perhaps certain forms of very valuable freight. In all of these respects it seems to promise the creation of a new super-luxurious transport field of its own rather than to threaten a very serious invasion of the fields already occupied by the railroads. Nor should we lose sight of the fact, which is now pretty clearly established by the experience in Europe today, that in the stage of evolution now reached or in sight, extensive air transport is, at present, only possible on a commercial basis when aided by government subsidies or guaranties.

No Considerable Extension of Line Mileage Probable

Considering these three physical factors which I have indicated—namely, the present broad development of the railroads, and the probable effect upon them of motor and air transportation—I am therefore inclined to think that we are unlikely ever again to see any very rapid increase in the existing line mileage of the railroads. In fact, as I mentioned a few moments ago, extension of line mileage has been practically at a standstill for some years past.

Opportunities for Improvement and Betterment

This, however, is very far from meaning that we are approaching the end of railroad betterment and improvement. On the contrary, the field for advancement is to my mind greater than ever before. There is almost unlimited opportunity, and not only opportunity but need, for the continuance, on a greatly augmented scale, of the intensive development of the established lines. Millions of dollars will be needed, and are needed now, to provide the additional double tracking which nearly all systems require, and the quadruple, sextuple, and even higher forms of multiple tracking which to a constantly greater extent our important trunk lines are certain to need in order to keep up with increased production and with the constantly growing commercial activity and more frequent travel which are so characteristic of the business and social life of the country.

New, enlarged, and improved passenger stations are needed at this very day for many communities, both large and small. Many more will be required as time goes on. Curvature should be eliminated on important lines, bridges strengthened, grades lowered, and the work of separating grade at highway crossings be continued with a fair and reasonable division of the expense between the railroad affected and the community benefited. Although locomotives have been vastly improved, particularly in the last two decades, there is still a great field for further progress in this direction, and equally so in the further adoption of improved operating methods, which will enable us to make better use, at all times, of the available motive power. Last, but by no means least, there is the great and, as yet, little touched field of electrification, which in future years should, and will, be extended where conditions economically warrant that step.

It is, of course, needless for me to say that no program of progress can be realized at all without adequate funds, and that brings us back once more to the fundamental problem of railroading, which is that of stabilizing credit by raising the investment return permanently to an adequate level, which of course means increasing the net earnings. In the last fifteen years the return on property investment in the railroads of the United States has barely averaged 4 per cent. Only once has it risen to 6 per cent and on one or two occasions it has very nearly approached the vanishing point The great task and duty of regulation is to improve that condition, and our regulative authorities, if they are to have the courage to do so, and if they are to pursue a constructive course, require absolutely the strong backing of public opinion, and particularly of our business men and of the professions identified with transportation. It is no more than just to add that the regulative authorities, on their side, quite properly require from railroad managements continued efficient operation, and the rigid elimination of all waste. Happily, as I stated a few moments ago, I believe the situation warrants genuine optimism, and is far more encouraging than at any time in our generation.

The Future Physical Development of American Railroads

To summarize, therefore, my view is that the future physical development of American railroads will center upon a vast program of intensive improvements and betterments, with ultimately what may amount in some instances to almost a complete rebuilding of the existing plant, while extensions into territory not yet occupied by rail lines will go ahead very slowly, if at all.

Communication between the existing rail lines, as well as in territory not now served by them and not warranting rail extensions, will, I believe, be chiefly taken care of by motor cars on improved public highways.

Lastly, for those who have the price to pay, air transport will establish new straight lines from city to city, and doubtless across both the oceans which wash our shores.

Incidentally, I expect to see the last vestiges of commercial traffic on inland canals disappear from the United States, and all fanciful projects for canalizing rivers not naturally navigable relegated to oblivion. In a nation so plentifully supplied with railroads with their exceptionally cheap rates, for the service rendered, having practically a motor car for every family, and holding the pioneer honors in the art of flying, such a slow, cumbersome, easily interrupted, expensive, and inefficient method of transportation as is afforded by artificially constructed and maintained inland waterways—often closed for half the year by climatic conditions—can no longer have a real place.

Coördinating the various forms of our permanent agencies of transportation will be a gigantic task, but one not impossible of solution, especially in light of the new views now prevailing, which, unfortunately, have completely reversed the old theories once held as to the danger of permitting mergers or consolidations. In the railroad field, consolidations are now declared by law to be the national policy, where once they were largely forbidden. At one period we worshipped enforced competition as a fetish. Now we know that competition has only such value as practical experience may demonstrate, and is now largely confined to service.

Personally, I believe that a proper degree of competition is and will continue to be necessary among the railroads themselves, as well as between the railroads and the other agencies of transporta-

tion. This is a requisite if we are to keep the spirit of initiative and invention up to the highest pitch, and so render possible the continued progress of the transportation art. But competition which merely means useless duplication of facilities and service is a waste, the cost of which is ultimately borne by all users of the service, in addition to which investors in the long run usually suffer partial or total destruction of their capital.

I have publicly stated many times, and take this occasion again to say, that I am thoroughly in sympathy with the general proposition that it is desirable to concentrate our American railroads into a comparatively small number of large systems. As a matter of fact, in my experience on the Pennsylvania Railroad I have been engaged in effecting many mergers and consolidations, extending over a long period of time. By them we have throughout the life of the company succeeded in bringing down the total number of active companies which have in the past aggregated something over 600 to approximately 70. So, you may see that our company and I personally have advocated and practiced consolidation for many years, in so far as state and national laws permitted.

I am, however, definitely opposed to any movement looking toward putting consolidations into effect on an arbitrary or forced basis; or to any thought that intelligently planned, conservatively financed, and properly managed railroads should be penalized by being made to bear the burden of losses incurred in the operation of lines which, perhaps, should never have been constructed at all. In this connection, I heartily commend the views of President Coolidge and the stand which he has taken upon this subject and has made known through the columns of the newspapers, which is briefly, that voluntary and permissive consolidations, with the encouragement and assistance of the governmental authorities, and coupled with proper safeguards for the public interest, constitute the sound and proper method of procedure.

In conclusion, it has been a pleasure and honor to have this opportunity of assisting in welcoming the members of your great profession to Altoona, which I think I may say is the world's leading center of mechanical engineering, as applied to the industry of transportation. Altoona, ever since it was founded 73 years ago by our company, has symbolized mechanical progress on the Pennsylvania Railroad, and I may fairly add, to very large extent on the railroads in general.

This city and our company are today the hosts of your association and profession. Engineering and transportation in general have alike made vast forward strides since you last met here, 42 years ago. We are proud, indeed, to have you as our guests again, and trust that this occasion may be long and pleasantly remembered in your organization's history and may serve to cement still more closely the bonds which have always existed between the profession of engineering and the Pennsylvania and other American railroads.

An Investigation of the Efficiency and Durability of Spur Gears

THE work described in the publication under consideration was carried out chiefly on a gear-testing machine designed by Wilfred Lewis, Mem. A.S.M.E. The object of the investigation was to secure reliable information on the efficiency of spur gearing and data on the change and the rate of change in the profiles of spur-gear teeth subjected to wear.

Among other matters of interest described is a tooth-profile indicator specially developed for use in durability tests. It is stated that the sensitivity of the instrument was such that the indicating fingers responded to the slightest variation in the tooth profiles. Since only one side of a tooth was tested at a time the readings on the other side remained constant during the entire durability test, and hence served as a check on the adjustment of the instrument. Measurements of the same tooth repeated even after an interval of two or three months, during which time many other measurements were made, showed an agreement within 0.0008 in. Because of lack of space only the conclusions arrived at can be reproduced here.

CONCLUSIONS ON EFFICIENCY (CONSIDERING TOOTH FRICTION ONLY)

1 The efficiency of unhardened gears is practically independent of the quantity of oil used for lubrication, provided the quantity is sufficient to prevent heating and cutting.

2 The efficiency is independent of the speed within the range covered by this investigation, namely, a pitch-line speed of 60 to 1500 ft. per min.

3 The efficiency does not appear to be influenced by the obliquity of action.

4 For all practical purposes the efficiency is independent of the load transmitted. The value of 99 per cent is suggested for use in computations dealing with the efficiency of gears cut in accordance with good commercial practice.

5 Condition of tooth surface is the most important of the factors which affect the efficiency of unhardened gears. Gears with rough tooth surfaces are less efficient than those in which the tooth surfaces have become glazed, but the difference in efficiency is not as great as has been commonly assumed.

6 When all other conditions are the same, greater sliding action causes the longer-addendum gears to have a slightly lower efficiency than the shorter-addendum gears. On the other hand, the vibration of the longer-addendum gears may, for certain ratios, be so much less than that of the shorter-addendum gears, as to result in a slightly higher efficiency of the long addendum gears.

7 The difference in efficiency of the several standard tooth forms in common use is so small as to exercise no controlling influence on the tooth form to be recommended or adopted for any purpose.

CONCLUSIONS ON DURABILITY

1 Unhardened steel pinion teeth quickly wear to outlines other than true involutes, regardless of load, speed, and lubrication; after this occurs wear practically ceases, or is greatly retarded under ordinary operating conditions.

2 The teeth of cast-iron gears meshing with steel pinions fail by crushing of the material in the region of the pitch line.

3 The teeth of unhardened steel pinions meshing with cast-iron gears fail by abrasive action which wears off the face of the tooth and hollows out the flank, resulting, in general, in a final outline of double curvature.

4 Lubrication is a very important factor in the life of unhardened gears. Although the quantity of lubricant does not materially affect the final change in tooth outline due to wear, it postpones the beginning of the rapid wear of both pinion and gear and thus greatly prolongs the life of the gears.

5 Under the same conditions of load, speed, and lubrication the wear increases with greater sliding action, but these tests indicate that the amount of sliding and the amount of wear are not necessarily proportional.

6 Combinations of factors or conditions which cause excessive vibration have a detrimental effect on the durability of gear teeth.

7 Surface pressure is the most important of the factors which affect durability. Apparently for any pair of gears there is a critical surface pressure, governed by the properties of the materials, above which the life of gears is short and below which the gears will run indefinitely without appreciable wear.

8 For unhardened gears under constant load and free from impact loads the allowable tooth loads based on the Lewis formula for strength are in excess of those permissible for satisfactory durability.

9 Unhardened gears for constant load transmission should be designed on a basis of durability rather than of strength. A gear tooth that is durable for a given load will, in general, be amply strong.

In appendices are given formulas for computing efficiency and a historical review (with bibliography) of the principal investigations of the efficiency, durability, and strength of toothed gears.—C. W. Ham, Mem. A.S.M.E., Assistant Professor of Machine Design, and J. W. Huckert, Research Graduate Assistant, University of Illinois, in *University of Illinois Bulletin*, vol. 22, no. 47, July 20, 1925 (which is also Bulletin 149 of the Engineering Experiment Station, University of Illinois), 92 pp., 42 figs.

The Rustproofing of Materials

Rust Resistance of Copper-Bearing Steel and Savings Possible through Its Use for Car Construction—Protection of Steelwork by Painting—Baking Process of Drying Painted Cars

By M. E. McDONNELL,[1] ALTOONA, PA.

THE Secretary of Commerce, Mr. Herbert Hoover, is making a notable fight against waste, and for the conservation of our resources. In his preface to a book entitled, Waste in Industry, he states that "Present waste in industry represents a huge deduction from the goods and services we might all enjoy." It is believed that that statement may be safely accepted as an axiom; and it would accordingly follow that while the unnecessary or avoidable destruction of a structure may represent a serious loss to its owner, the loss goes further and affects all of us due to its encroachment on our national resources. The extent of the damage due to rust and corrosion of metals in this country is receiving wide recognition. In his address before the American Society for Testing Materials in June of this year, President F. M. Farmer estimated that in this country we are losing $300,000,000 annually due to this cause. At the spring meeting of the American Chemical Society an entire session was devoted to a symposium on corrosion of metals. The American Society for Testing Materials, the American Electrochemical Society, and other organizations have active committees assigned to the study of corrosion, and much has been accomplished.

Through the medium of *The Iron Age*, Dr. H. W. Gillett of the Bureau of Standards informs us that eight per cent of the funds allotted to the Bureau is expended on research, the object of which is to determine the causes and means of prevention of corrosion.

The research department of the Pennsylvania Railroad System has been actively studying rust and corrosion for years, and the protective measures suggested by experiment have been applied on a broad scale by the management. This would therefore appear to be an opportune time to review what has been accomplished.

Losses Chargeable to Corrosion

In considering the effect of the destruction of steel due to rust, it should be remembered that the loss of 1000 tons of steel gives a depletion to our national resources equivalent to 2000 tons of ore, 4000 tons of coal, more than 500 tons of limestone, together with losses of magnesite, chromite, manganese, etc.; and that even now most of our manganese ore and some iron ore are being imported. The labor required in assembling these involves a large number of "man-hours." Also, consideration of the loss due to rust cannot be confined to our material welfare. Rust and corrosion frequently impair the strength of structures, thereby endangering lives. The recent loss of some fifty lives on the S. S. *Mackinac* was reported to have been due to a corroded piece of steel. Many other similar illustrations could be given.

In nature the metals usually employed for structural purposes normally occur as compounds such as oxides or sulphides, and not in the free metallic state. Through metallurgical operations of an artificial character, involving high temperatures and reducing atmospheres, the metals are obtained. Under the influence of nature these metals normally revert to the same type of compounds from which they originated. The structural materials used by the engineer are not in equilibrium, and all of them are under a potential pressure to revert to a balanced state. This change of state is commonly termed "corrosion." The mechanical engineer must ever bear in mind this corrosive force in the selection of his materials, as well as their subsequent protection.

Rust Resistance of Copper-Bearing Steel

Engineers now know that commercial steel on the market varies greatly in its tendency to rust. As previously stated, many tests have been made by the research department for the purpose of developing resistance to corrosion. Fig. 1 shows the main Test Department Building which has been provided by the management of the Pennsylvania Railroad System. Fig. 2 shows a laboratory equipped with apparatus for hydrogen ion determination, which is now being used in studies planned in search for means for preventing the pitting and corrosion of locomotive boiler tubes. On August 1, 1912, fifteen weighed commercial steel sheets were exposed on the roof of the Locomotive Test Plant in Altoona. These sheets were carefully analyzed, and on February 13, 1913, after six and one-half months' exposure, they were wire-brushed and reweighed. It was found that the rate of rusting varied much, the minimum loss being 30.12 grams and the maximum loss 84 grams on the two sides of a square foot of the various sheets undergoing test. The results are shown in Fig. 3, and it is to be especially noted that the sheets showing the lowest losses in weight contained copper. The curve on the plot shows the copper content. This hitherto unpublished

Fig. 1 Physical and Chemical Laboratories, Pennsylvania Railroad System, Altoona, Pa.

chart was prepared in 1915, and it was largely instrumental in bringing about the decision of the Pennsylvania management to adopt copper-bearing steel for car roofs.

A more vivid illustration of the variation of the durability of steel is shown by three photographs showing the progress of failure of a lot of 258 No. 16 gage and 230 No. 22 gage sheets of plain-carbon and copper-bearing steel which were exposed for test purposes in Pittsburgh on December 12, 1916, by a committee of the American Society for Testing Materials of which the author had the pleasure of being a member. Fig. 4 shows these test racks after 16 months' exposure of the sheets. The records show that 30 No. 22 gage plain-carbon sheets failed in sixteen months, while no copper-bearing sheets failed in that length of time. Fig. 5 shows the same racks after 28 months' exposure, and it will be observed that during that period some of the panels had turned entirely to rust. The number of No. 22 gage plain-carbon failures had increased to 77, this being more than 92 per cent of this class of sheets. During the same period only 6 No. 22 gage copper-bearing sheets had failed, and it should be stated that none of them contained manganese. Fig. 6 shows the racks after 75 months' exposure. Many gaps are shown from which entire sheets of plain-carbon steel had disappeared as rust, in which the oxide may have been associated with some sulphate of iron. At the end of that period every plain-carbon-steel sheet had failed, while 13 copper-bearing No. 22 gage sheets were free from holes or ragged edges. Also, at the end of that period 102 of the 126 No. 16 gage plain-carbon-steel sheets had failed,

[1] Chief Chemist, Pennsylvania Railroad System.

Presented at the Altoona Regional Meeting of The American Society of Mechanical Engineers, Altoona, Pa., October 7, 1925.

while there was not a single failure among the 132 No. 16 gage copper-bearing-steel sheets. The average life of the copper-bearing No. 22 gage sheets on this test was over 49 months, while the average life of the 84 plain-carbon-steel sheets was 23 months. The results of this test are fully recorded in the annual report of the American Society for Testing Materials for 1923. The author's tests of 1913, together with the obvious outcome of the A.S.T.M.

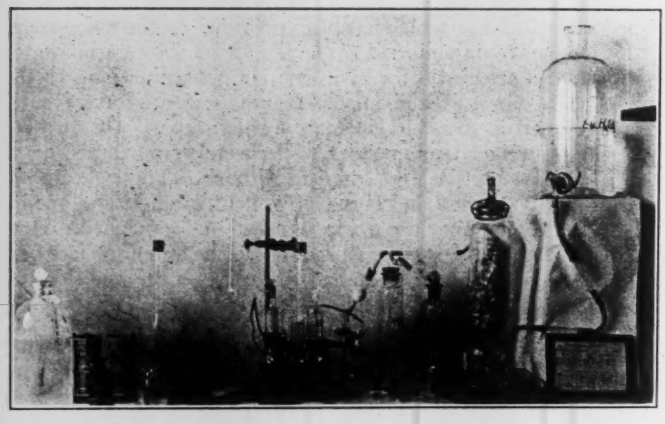


FIG. 2 LABORATORY EQUIPMENT FOR HYDROGEN ION DETERMINATION

Pittsburgh test, led the Pennsylvania management in 1919 to adopt copper-bearing steel for all sheet steel to be used in cars.

A question which doubtless comes to mind at this point is, what will be the effect of this change in the composition of steel which was made in 1919? The author does not know the answer to this question. The destruction of a car is not due entirely to rust. There is some destruction due to abrasion and wear, also to the action of acid such as may develop when wet coal is stored in cars. Car sheets are sometimes destroyed mechanically in the removal of wet or frozen lading. The tests cited indicate that the steel now being used is more than twice as resistant to atmospheric corrosion as the steel formerly used. The various factors have been carefully considered, and we shall expect a 50 per cent increase in life, due to the change from plain-carbon to copper-bearing steel.

SAVINGS POSSIBLE BY USING COPPER-BEARING STEEL FOR CAR CONSTRUCTION

The Pennsylvania Railroad System has some 269,000 freight cars, of which 266,588 are of steel construction, 167,398 of these being steel throughout. These steel cars are of different weights and designs. There are approximately 32,000 all-steel box cars, on which the rate of depreciation due to rust is not yet definitely known. They can be protected against rust by means of paint coatings to a much greater extent than an open car which is used for rough freight, and the roof protects the inner side of the plates. No attempt will therefore be made to estimate the value of copper-bearing steel for this class of cars.

There are 135,525 open all-steel freight cars on which a large amount of maintenance data is available. After these cars have been in service for a number of years, the sheets become thin and holes develop. When they have reached this condition it is possible to patch the failed sheets, but if this is done, the surrounding sheets are likely to fail and the cost of this kind of maintenance is likely to become excessive. When failure of sheets begins to occur due to rust, it is better to apply Class 1 repairs, which involves cutting all rivets, removing sheets, and rebuilding the car body. The age of the car at which this class of repairs is called for varies some, depending on design and service. The standard four-hopper coal car designated H21a now requires this attention after ten years of service. The two-hopper coal car of the G1 type is of slightly more rigid construction than the four-hopper car. It will run somewhat longer than ten years prior to Class 1 repairs, possibly as long as twelve years. On the other hand, on the gondola car of the Gsd type, the sheets fail after approximately eight years' service. This rapid failure is partly due to the fact that this car does not drain and dirt accumulates in it, which, when wet, accelerates rusting. A statement has been prepared (Table 1) showing the cost of maintaining car bodies, in which the calculations are based on an average length of life for the plain-carbon steel, which is ten years between Class 1 repairs. The results in columns 1 to 4, inclusive, are from actual data, while those in column 5 are averages of the other four. The estimates of materials and cost of maintenance for copper-bearing-steel cars represent an anticipated 50 per cent greater durability than for the plain-carbon-steel cars, or a 15-year service prior to Class 1 repairs. No allowance is made for painting the cars or for any repairs to the underframes, trucks, or brake equipment. The increased cost of cars due to the use of copper-bearing steel is shown. This is obtained by applying the differential of $3 per ton which has prevailed between plain-carbon and copper-bearing steel plates. This amount may diminish when the amount of copper-bearing scrap now being used by the steel makers is increased. In making the calculation, allowance is made for losses in fabrication caused by shearing and punching. It is shown that the 135,523 cars involved would have cost $2,509,295.86 more if they had been made of copper-bearing steel. The statement shows the amount of new finished plates, shapes, and rivets, as well as the cost of material and labor, including shop expenses, required to dismantle and rebuild car bodies. If these cars are given Class 1 repairs over a period of ten years, using plain-carbon-steel sheets, the annual cost under present

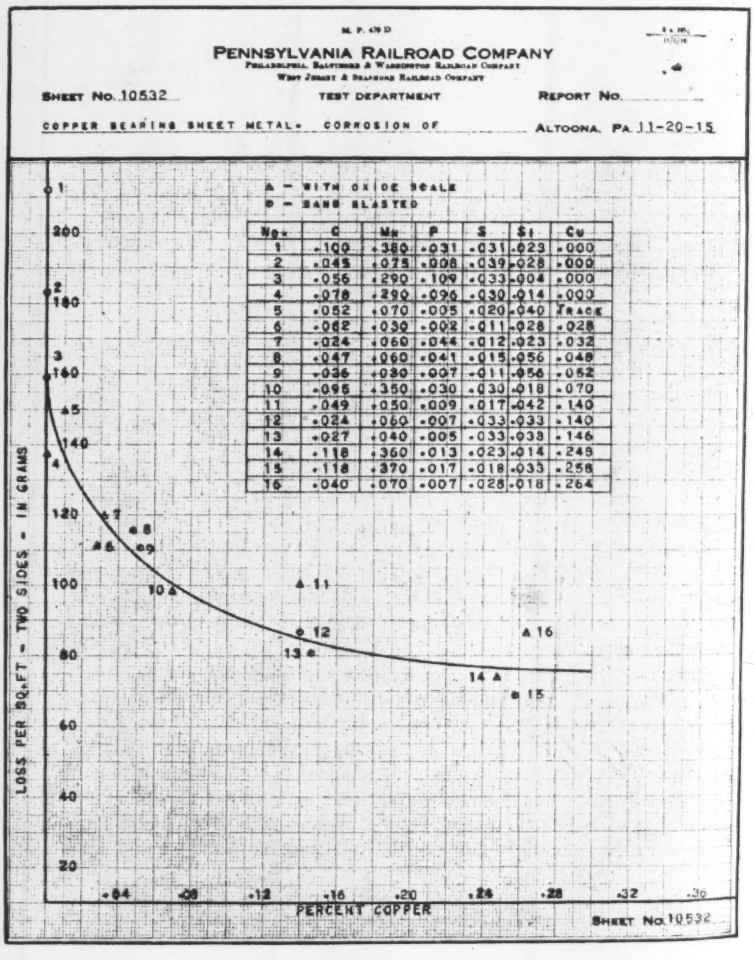


FIG. 3 VARIATIONS IN RATE OF CORROSION OF STEEL

market conditions amounts to approximately $5,069,112.03. From the experimental data at hand to date, it may be assumed that with the use of copper-bearing steel the interval between Class 1 repairs will be extended over a 15-year period, and repairing with copper-bearing-steel sheets reduces this cost to approximately $3,473,710.30. This represents an annual saving of $1,595,401.73, and an annual reduction of 22,385 tons in the amount of new steel required.

Another important advantage gained by the reduced number of

shopping periods is the increased number of days that the cars are serviceable. No attempt has been made to capitalize this advantage, and the gain does not show in Table 1. Transportation officers and shippers well know the serious consequences which may result from lack of the necessary cars to convey the commodities of a community.

Numerous other economic factors enter, which are very patent to students of political economy. The reduction in the amount of steel required, making due allowance for scrap returned to open-hearth furnaces, reduces the ore consumption some 20,000 tons and the coal consumption not far from 80,000 tons. Among numerous other items conserved are chromite, magnesite, compressed oxygen gas, acetylene, fuel oil, etc. Of even greater importance is the conservation of labor, which can be diverted to other channels. They are all contributory factors to what Secretary Hoover includes under "deduction from the goods and services we might all enjoy."

Influence of Copper Content on Corrosion of Steel

Those who have not given this subject special study may wonder why copper is a protective measure and what connection it can have with protective coatings. Those who have had practical experience with steel may have noticed that ordinary steel forms a light brown, loose rust on oxidation, while copper-bearing steel oxidizes on the surface to a dense, dark brown adherent coating, and this is believed to act as a protective measure against rapid destruction. Other similar examples of oxidation-protective coatings are well known to the engineer. A bright copper roof has a rapid initial rate of corrosion, but the dark-green coating formed on exposure soon retards further progress. Bronze has a rapid initial rate of oxidation, but the beautiful adherent coating arrests destruction. Zinc is very readily corroded, but the coating formed protects the base metal. In the selection of materials for engineering structures, the character of the coating formed on oxidation deserves careful consideration. Rust-proofing coating may therefore be induced by the use of a small percentage of an element or ingredient which on oxidation produces a protecting film.

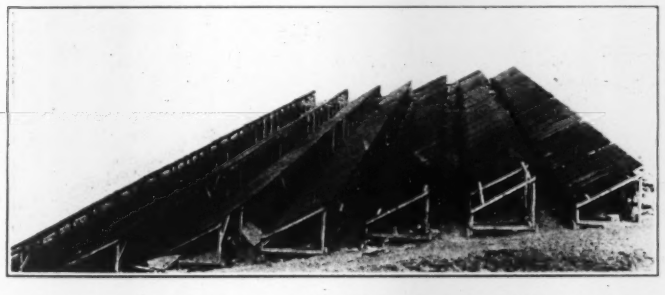


FIG. 4 PITTSBURGH TEST RACK—SHEETS EXPOSED DEC. 12, 1916; PHOTOGRAPHED APRIL 18, 1918

In the tests cited, special reference has been made to the profound influence of copper. It is not to be inferred from this, however, that other elements present in small quantities do not influence the rate of rusting. In the absence of copper, a high sulphur content stimulates corrosion. In the presence of copper, a small percentage of manganese improves the resistance of the protective coating formed on oxidation. The presence of chromium in steel increases its rust-resisting properties, and the so-called "stainless" steel which is now being used for cutlery and other small parts where great rust resistance is essential, contains approximately 13 per cent of chromium. However, various causes, such as the limited amount of chromium ore available, the high cost of its reduction, and the difficulties encountered in rolling stainless steel, will likely prevent its use for general structural purposes.

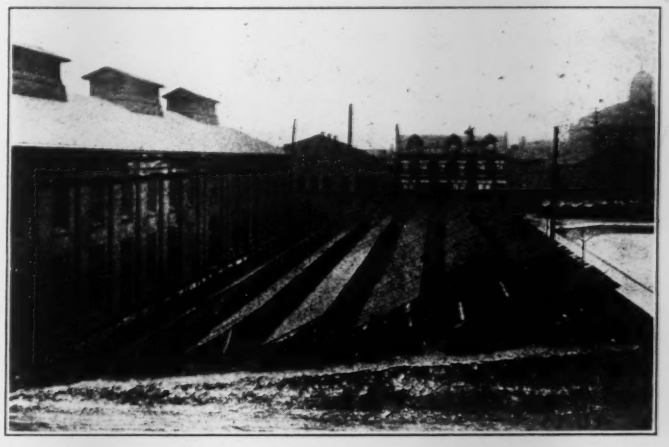


FIG. 5 PITTSBURGH TEST RACK—SHEETS EXPOSED DEC. 12, 1916; PHOTOGRAPHED APRIL 14, 1919

FIG. 6 PITTSBURGH TEST RACK—SHEETS EXPOSED DEC. 12, 1916; PHOTOGRAPHED MAR. 9, 1923

TABLE 1 COST OF STEEL-CAR MAINTENANCE AS AFFECTED BY RESISTANCE TO RUST

	Four-Hopper Coal Car Class H21a	Two-Hopper Coal Car Class Gla	Two-Hopper Coal Car Class Glc	Gondola Car Class Gsd	Miscellaneous Hopper and Gondola Cars Classes Gs, Gsa, G22, H22, etc.[1]
Number of cars in service	34,854	30,071	6,183	22,431	41,984
Light weight of car, lb.	50,700	39,500	39,000	40,700	....
Finished Plates and Shapes in a New Car:					
Weight, lb.	13,293	9,292	13,124	9,675	11,346
Cost, if plain-carbon steel	$ 354.79	$ 248.00	$ 350.28	$ 258.23	$ 302.83
Cost, if copper-bearing steel	377.26	263.71	372.46	274.58	322.00
Increased cost per car, using copper-bearing steel	22.47	15.71	22.18	16.35	19.17
Class 1 Repairs, Per Car Basis:					
Finished plates and shapes, lb.	10,670	7,461	11,081	7,830	9,261
Rivets, lb.	870	793	730	870	816
Gross cost of finished plates, shapes and rivets, including stores expenses:					
Plain-carbon steel	$ 309.80	$ 225.12	$ 317.74	$252.85	$ 276.37
Copper-bearing steel	327.04	237.18	335.65	265.50	29..34
Credit for scrap	38.47	27.51	39.70	29.00	33.67
Net { Plain-carbon steel	$ 271.33	$ 197.61	$ 278.77	$ 223.85	$242.70
Net { Copper-bearing steel	288.57	209.67	295.95	236.50	257.67
Labor directly applicable	120.35	90.10	81.48	91.30	95.81
Shop expenses directly applicable	54.34	40.86	36.59	40.99	43.20
Total { Plain-carbon steel	$ 446.02	$ 328.57	$ 396.84	$ 356.14	$ 381.71
Total { Copper-bearing steel	463.26	340.63	414.02	368.79	396.68
Annual Cost per Car of Class 1 Repairs to Finished Plates and Shapes:					
Plain-carbon-steel car, ten-year renewal	$ 44.60	$ 32.86	$ 39.68	$ 35.61	$ 38.17
Copper-bearing-steel car, fifteen-year renewal	30.89	22.71	27.60	24.59	26.45
Saving per car per year	$ 13.71	$ 10.15	$ 12.08	$ 11.02	$ 11.72
Total anticipated saving	$477,848.34	$305,220.65	$74,690.64	$247,189.62	$491,524.48
Plain carbon steel and rivets, lb. per car per year	1,154	827	1,181	870	1,008
Copper-bearing steel and rivets, lb. per car per year	769	551	788	580	672
New steel saved, lb. per car per year	385	276	393	290	336
New steel saved per year, tons	6,709	4,155	1,216	3,252	7,053
Difference in cost between carbon and copper-bearing steel for the 135,523 new cars					$2,509,295.86
Total anticipated saving per year for the 135,523 cars, due to slower rusting of copper-bearing steel					$1,595,401.73 / 22,385 tons of steel

[1] All figures in this column are average estimates.

Designing engineers are therefore urged not to rely solely on physical tests when adopting specifications for steel to be purchased for structural purposes, if the material purchased is for other than temporary structures. Chemical engineers are making active search for new additions which will advance the state of the art to a still greater degree of efficiency.

Pieces of steel which are identical in chemical and physical properties disintegrate at varying rates if subjected to different influences. In the absence of oxygen or free mineral acids, the rate of destruction is slow. In the presence of oxygen, some natural waters quickly form a protective coating due to the presence of salts which they carry in solution. Pure water does not form these coatings. New York City water is relatively pure, and it has a persistent action on metal pipes. Removal of the oxygen largely eliminates

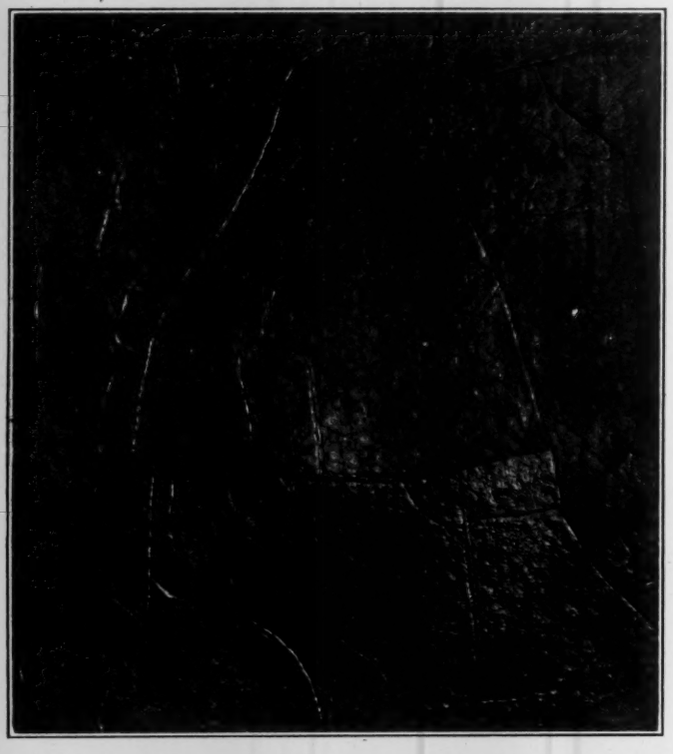


FIG. 7 SHOWING HOW AN AIR-DRIED INTERIOR CAR FINISH CONTAINING DRIERS AND GUMS IN EXCESS FAILS

this action. The water supplied to the Hotel Pennsylvania in New York City is treated with a small quantity of sodium silicate. This forms an insoluble silicate coating on the interior of the water pipes, which protects them. Incidentally, this prevents the water in the pipes from becoming "rusty," a condition which might give rise to complaints by the guests of the hotel.

In the application of laboratory tests or experimental installations the results obtained apply only to the conditions under which the tests were made. It has not to our knowledge been definitely proven that copper-bearing steel has any merit in forming a protective coating on submerged structures, or for boiler tubes. Pieces of all of the sheets exposed on the Pittsburgh test rack were immersed in mine water at Calumet, Pa., in the Potomac River at Washington, and in the Severn at Annapolis, but in these tests no effect could be attributed to copper. Monel metal offers marked resistance to corrosion under normal atmospheric tests, and the engineer who specified monel metal for the steam yacht *Sea Gull* probably possessed a mass of data which led him to believe that he was adopting "the last word" in engineering efficiency. He failed, however, to reckon with electrolytic conditions, which did not pertain to the examples coming within his previous experience.

PROTECTION OF STEELWORK BY PAINTING

The duties of the engineer do not end with the purchase of the best commercial structural metals now available. Steel for important structures, such as bridges, should receive a shop coat of good paint prior to its shipment from the factory, and for this purpose red lead is still a favorite. Steel for rolling stock which is not painted should be stored in a dry place until it can be assembled, and if this is not feasible, a protective coat of some kind should be applied. The method employed by the Pennsylvania Railroad System and, it is believed, by numerous other concerns, is to apply a petroleum-oil coating which contains resins and wax, these constituents being added to the oil for the purpose of imparting adhesive properties. On withdrawal of the steel from storage for use, the temporary protective coating is removed. After construction, exposed steel parts should be well painted. Many technical organizations, including the company's research staff, have devoted a large amount of work toward determining the most durable paints which can be applied. Experience teaches us that the best air-drying paints are those which contain suitable pigments, good oil, and a minimum amount of drying constituents, such as japan drier. Paints made in accordance with these precepts dry slowly, and under conditions prevailing in industrial centers the coating is likely to become contaminated with dirt before the paint film is set. Furthermore, the use of such paints on rolling stock keeps it in the shops and out of service. Painters know that an excessive amount of driers shortens the life of paint. The action of the driers does not cease when the paint film has become dry, but continues after the purpose for which they were added has been accomplished. This leads to premature destruction of the paint film, and while the fact is generally known, it has been confirmed by the P.R.R. by panel tests. Notwithstanding the known detrimental effect of these driers, painters sometimes use them in excess in order to get the equipment out of the shops in the shortest time possible. Fig. 7 shows how an interior air-dried passenger-car coating containing excessive driers and gums fails. Numerous tests were made to solve

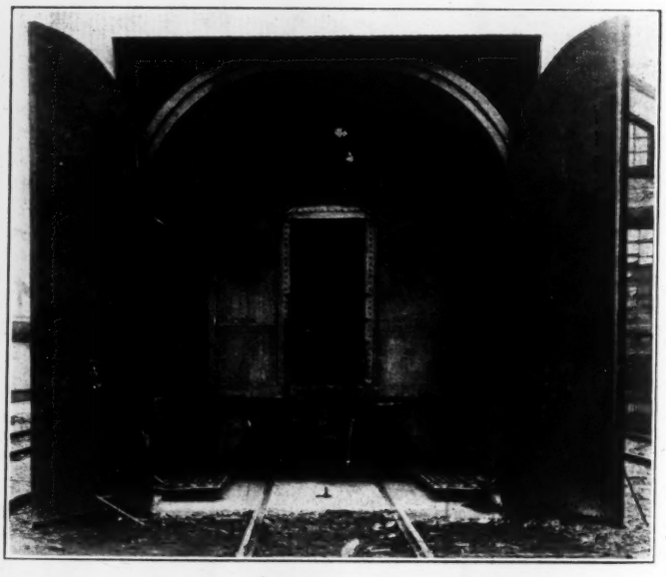


FIG. 8 BAKING OVEN USED IN 1913

this difficulty, and it was found that by means of heat, paint films could be dried without the use of artificial driers.

BAKING OVENS FOR DRYING PAINTED CARS

A laboratory baking oven was constructed and a large number of painted panels prepared, using different compositions. These panels were exposed on a test rack in comparison with panels painted with the usual air-drying paints and varnishes in use. The panel tests showed that the baked coatings had greater durability than the air-dried ones. The difference was so decided that a baking oven for passenger cars was constructed at the Altoona paint shops. The first car painted by the baking process was finished in January, 1913, and the paint was in good condition after forty months of service, during which time the car did not receive any applications of color or varnish. Another car painted by the baking system in November, 1913, was found to be in excellent condition September 22, 1925. During this period the car was in the shops for baking class repairs at three different periods. No

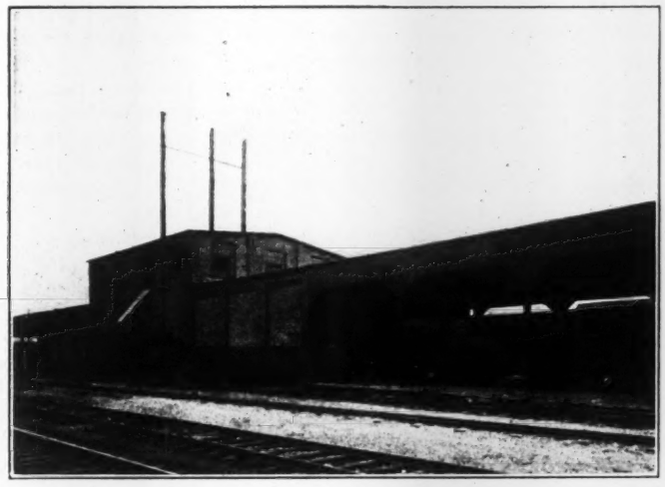


FIG. 9 BAKING OVENS AT THE PITCAIRN SHOPS—BUILT IN 1923

FIG. 10 FANS AND HOT-AIR FURNACE, PITCAIRN BAKING OVENS

color or yarnish has been applied to it since June, 1923. These class painting repairs consisted of one coat of color and two coats of varnish to the exterior, and one coat of varnish to the interior. Cars painted by the baking process require class repainting every thirty-six months, while those painted by the air-drying process receive class repainting at intervals of eighteen months. It requires fourteen working days to paint a new passenger car by the air-drying process, while by the baking process all of the painting operations can be applied in six working days. The baking process reduces the time which passenger cars must be retained in the shops on account of painting operations more than 60 per cent. There are now three baking ovens in service, one of which is shown in Fig. 8. They are 90 ft. 3 in. long, and will accommodate the largest cars in service. They are equipped with ventilators and are heated by steam under a pressure of 125 to 150 lb. per sq. in. Each oven contains 2000 sq. ft. of radiating surface. Temperatures of 250 to 260 deg. fahr. are obtainable. Baking paints, also the priming and surfacing coats for passenger-equipment cars, are dried in three hours at these temperatures. Varnish and light-colored enamels used for the final coats on passenger cars darken at high temperatures. In practice they are dried in three hours at temperatures of 150 to 160 deg. fahr.

The coatings as applied to passenger cars by this method have been found to be very durable. All steel passenger cars from which it is necessary to remove the old paint are sent to Altoona to be repainted by the baking process. All cars painted by the baking process are marked, and when they require class painting repairs, the additional colors and varnishes applied are dried in these ovens.

In addition to the baking of paint coatings on passenger cars, the method was applied to some 1100 freight cars in 1913. These were coal cars which were in need of paint. They were given one coat of baking paint in which different oils were used, including menhaden fish, corn, cottonseed and soya bean oils, with linseed and china wood oils. None of the paints used contained artificial driers. The paint on these cars was dried in an oven at South Amboy which was used for thawing cars of frozen coal which were to be dumped into boats. The temperature obtainable was 185 to 195 deg. fahr., and it required six or seven hours to dry the paint. The cars were repeatedly inspected for about five years, and the results obtained were so satisfactory that baking ovens were authorized and built at Pitcairn Shops in 1923.

Fig. 9 shows the Pitcairn installation, the ovens of which, it will be seen, are different from the Altoona ovens previously described. There are three units over parallel tracks, each of which will accommodate three freight cars at one time. Each unit is provided with a coal-fired hot-air furnace containing forty tubes through which air is circulated by means of fans. The air to be heated is drawn from the top of the baking oven, passed through the tubes of the hot-air furnace, and forced back to the bottom of the ovens through suitable ducts. The circulation of the air from the ovens through the furnace is continuous. However, provision is made

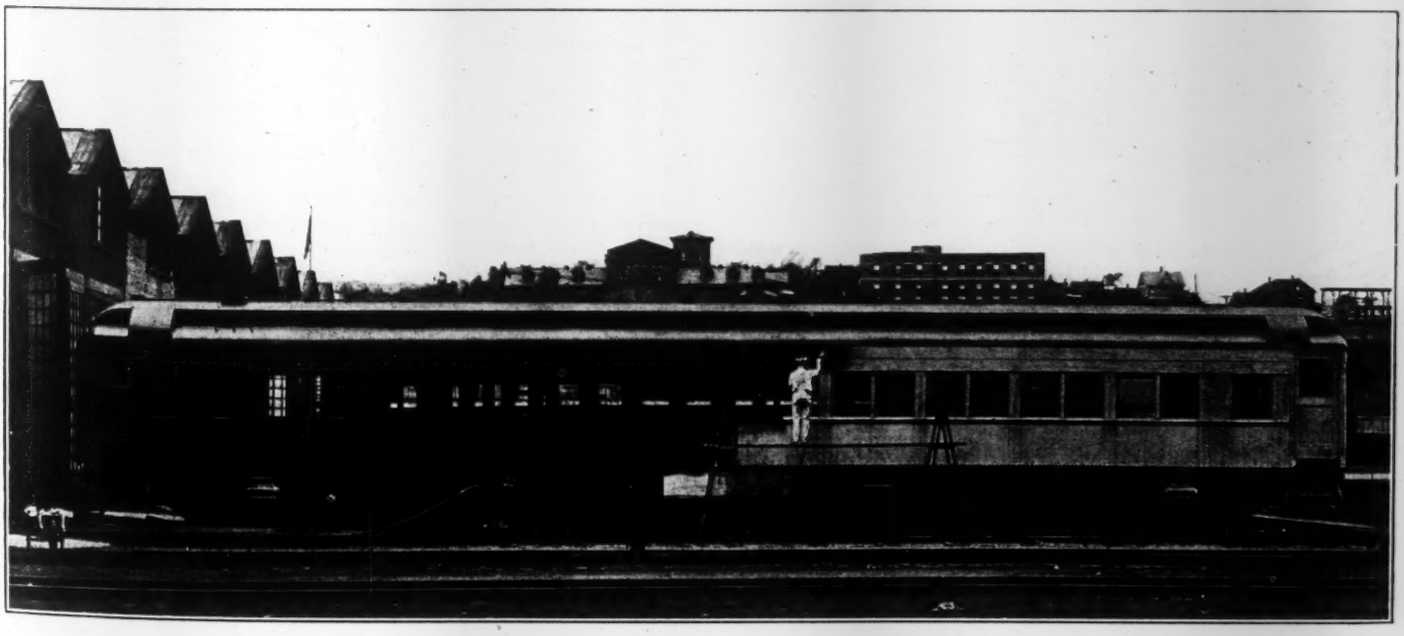


FIG. 11 APPLYING TUSCAN RED LACQUER FINISH TO CAR WITH SPRAYING EQUIPMENT

for a little ventilation. A vent is provided for discharging some hot air from the top of the ovens to the atmosphere, and this loss is replenished with fresh air which is drawn through an intake into the plenum chamber of the fans. Fig. 10 shows the furnace, the fans, and the motors which operate them. These ovens permit the control of any desired temperature up to 350 deg. fahr. The time required to dry the paint coating depends on the temperature. If the ovens are operated at 225 to 250 deg. fahr., the painted cars are allowed to remain in them for three hours; and at 275 deg. fahr., however, the time is reduced to two and one-half hours. Thirty-two cars are now painted daily at Pitcairn. Each car receives two coats of paint, both of which are baked for two and one-half to three hours, after which the cars are stenciled. The paint is sprayed on, and two shifts of painters perform the work. It is possible by this method to apply two coats of paint and stencil cars in one full working day after the cars are delivered to the paint shed, it being understood that due to two shifts of employees a working day is 16 hours. No artificial driers are added to the paint used, and a large percentage of menhaden fish oil is used in thinning the paste paint, which is ground in linseed oil. A small percentage of china wood oil is used, as our tests and experience indicate that this addition increases the water-resisting properties of the paint film. It is estimated that the time saved in painting cars by this operation is two days. In other words, the cars are available for two more days of revenue service. By this method of painting cars, unexpected delays due to failure of the paint to dry on account of bad weather conditions is not experienced. The method will likely be extended to other shops.

The extension of the baking system for passenger cars to other shops has been deferred pending the outcome of tests which are being conducted with the lacquer system which has recently revolutionized the method of painting automobiles. These lacquers contain a colloidal solution of nitrated cotton and varnish gum in a mixture of volatile solvents, which usually include ethyl acetate, anhydrous alcohol, benzene or toluene, and other volatile constituents. Pigments are incorporated with this solution to give the desired color and to make the resulting film durable. Lacquers of this type dry so quickly that they cannot be satisfactorily applied by means of a brush. They are applied with spray equipment. The Pennsylvania Railroad System now has five passenger-equipment cars and over a hundred locomotives which were finished with lacquer. Fig. 11 shows the operation of finishing the exterior of a passenger car with tuscan red lacquer. The exterior of a car can be coated without difficulty. There is some trouble in finishing the interior of cars due to the fact that several colors are used. If the headlining is to be light green, the body of the car bronze, and the base dark green, any one of the colors can be applied, but in the application of the adjoining color by means of a spray gun it is difficult not to spoil the first color applied. Another difficulty is encountered if it is desired to apply gold striping or lettering over a lacquered surface. The gold must be protected, and this is now done by pencil-varnishing it. This is a time-consuming operation. The lacquer coatings can be quickly applied, and successive coats can be sprayed at intervals of about thirty minutes. However, means will probably be devised for overcoming the present difficulties.

Lacquer coatings appear to be durable. A panel coated with lacquer and exposed twenty-one months was still in a good state of preservation at the end of that period. When properly made, applied, and polished these nitrocellulose-base coatings present a pleasing effect suggestive of refinement. Lacquer finishes can be cleaned more easily than the more familiar varnish coatings.

The progress in the development of rust preventives has been rapid during the past decade. The successful engineer must ever be alert to the end that his client may receive the benefit of the best information obtainable.

A Plea for Justice for the Small Boiler

FOR A NUMBER of years interest in the care and treatment of the small boiler has engaged more and more of my concern.

By the small boiler I mean the boiler used to generate steam in hotels, apartments, office buildings, laundries, creameries, and small industrial plants for heating or power purposes.

In this age of hungry interest in the gain of dollars, one would think the tremendous waste in fuel would influence owners of small boilers to take greater care of and give more attention to their boilers, since they virtually burn up their owner's money. It is not until one visualizes the situation, in full perspective, that we realize that the reason they fail to sense or feel this waste of fuel is because the cost of fuel is a minor item, compared to other expenditures, in their small plants. Operators of small boilers seem to feel that they are a sort of necessary evil and treat them as such, until they finally give up in despair and sometimes with an awful vengeance.

If a boiler is cleaned thoroughly and regularly, it will save in fuel double the cost of labor to clean it. If the tubes, shell, and furnace are kept free from soot and ashes, a like economy will be obtained. Steam and water escaping from the boiler through leaks result in waste of fuel by loss of steam and water containing heat units. Loss of fuel results from the excess water evaporated to make up for the leaks. Excess water brings in added scale and mud which are deposited on the heating surfaces of the boiler, thereby necessitating more fuel to penetrate the scale and sediment to heat the water.

Clean tubes, furnace, and combustion chamber once a week, whether they need it or not, and you'll find defects before they give serious trouble and loss.

A boiler properly installed will do its work with a minimum of active or immediate attention, and this fact leads the owner or employer to divert the activities of the fireman or engineer to other tasks that generally absorb all of this time, so in the end he gives less and less time to the care of the boiler until the neglect takes its usual toll of serious injury to the boiler and, maybe, of lives. The boiler will outlive you if you take good care of it.

Boilers in stationary work do not wear out, they do constantly waste out, though, and rust out. If the water deposits scale and sediment, treat the water before it enters the boiler and prevent burning of the plates, caused by excess scale or mud. You can not overheat a clean boiler, full to the water line of clean water. If the water of the district is corrosive, treat it to neutralize the acids that make it corrosive and save your boiler from their action.

Save all condensate possible and return it to the boiler, which is to say, save your money.

Take care of the fittings and appliances. Keep them operative. There are very few of them, and they are vitally necessary.

Keep the boiler clean inside and out. Keep the brickwork or housing tight and in good repair.

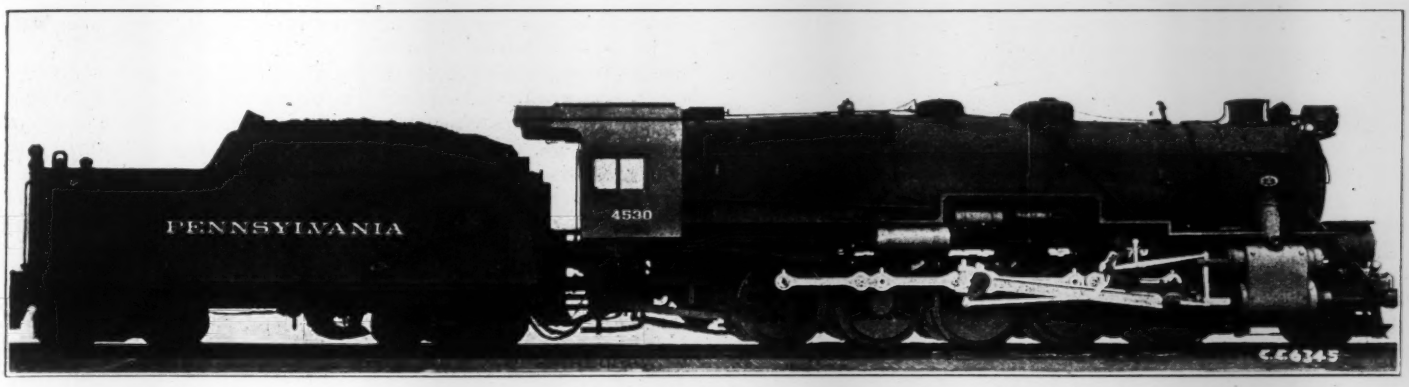
If the shell and tube surfaces inside of the boiler are clean, the heat of the fire penetrates and acts on the water as it should, causing rapid and economical generation of steam.

If the shell and tube surfaces of the outside of the boiler are clean, the heat of the fire acts directly on the metal and transmits the heat to the water through the steel, rapidly and economically. If the tubes and shell are sooty and dirty, you must use from ten to a hundred per cent more fuel to generate steam, and the same holds true if there is mud, scale, or sediment on the internal parts of the boiler.

If your furnace walls are cracked; if the fire doors are broken and hanging on by their eyebrows; if the combustion chamber is clogged with ashes, you are losing fuel by excess air. You are heating up great quantities of air that pass on through the furnace and up the stack, wasted.

These suggestions are all practical and can be applied in any boiler room. Above all, their regular practice is first and last a step to safety. A carefully attended boiler will not fail you or explode.

Set aside one day a month for a thorough cleaning of your boiler inside and out; see how much you will save in a year in fuel bills; how much more work you can get out of your boiler; and, how much longer it will last.—Benj. J. Parker, Boiler Inspector, Department of Safety, State of California, in *California Safety News*, June, 1925.



DECAPOD-TYPE LOCOMOTIVE 4530, CLASS H1s
(The latest development of the decapod type, which, since being introduced in 1918, has become the standard P.R.R. heavy freight locomotive. Tests are reported in Bulletin No. 32.)

The Locomotive Testing Plant and Its Influence On Steam-Locomotive Design

A Brief History of Locomotive Testing Plants and Their Contributions to the Development of the Locomotive—The Work Done by the Altoona Testing Plant—Contrasts of Size and Economy of Locomotives Made 20 Years Ago and at the Present Time

BY LAWFORD H. FRY,[1] BURNHAM, PA.

THE essentials of a locomotive testing plant are (*a*) that the locomotive driving wheels be supported on carrying wheels fitted with friction brakes to absorb the energy developed, and (*b*) that a dynamometer be provided to measure the drawbar pull. In the latest plants the friction brakes are of the Alden hydraulic disk type, which has proved to give satisfactory absorption of power and furnishes sensitive and accurate control of the speed.

On a modern testing plant it is possible to operate any given locomotive through its whole range of power. Within this range any desired combination of conditions, such as speed, cut-off, etc., can be selected and maintained constant during a run of an hour or more, and while running under such conditions measurements can be made with laboratory accuracy and completeness. It is this combination of constancy of conditions and accuracy of measurements which gives the results obtained on a testing plant their great advantage over road tests of locomotives. Road tests with a dynamometer car are valuable for checking and completing information obtained on the plant as to drawbar pull, but in all questions of steam production and consumption the plant results are far more authoritative. There is much detailed information which now controls locomotive design which it would have been practically impossible to obtain without a testing plant.

The locomotive testing plant is purely of American origin, and though the results of the tests made here have been widely studied, the author has been unable to find a record of any real testing plant installed outside of the United States. An appendix to the complete paper gives a fairly comprehensive bibliography of papers, articles, or other publications dealing with or deriving data from the various American plants. Locomotive-testing-plant history goes back to 1891, when the first plant was designed and erected at Purdue University under the supervision of Dr. W. F. M. Goss. A description of this plant was presented to the Society in 1892 by Dr. Goss. Locomotive testing plants have therefore been in use for more than 33 years, or one-third of the century during which steam locomotives have been employed in America.

Altogether there are records of six locomotive testing plants built and operated. The first Purdue plant, referred to above, was destroyed by fire in 1894, but was rebuilt and enlarged the same year. While it was out of commission the late Robert Quayle of the Chicago & Northwestern Railway installed a temporary plant at Kaukauna, Wis., to carry out tests for the A.R.M.M.A. committee on exhaust nozzles. In the following year, 1895, Mr. Quayle built a permanent plant at the Chicago & Northwestern Railway shops in Chicago. In 1899, as a result of the gift by the Baldwin Locomotive Works of a Vauclain compound locomotive, Columbia University built a testing plant under the direction of Prof. F. R. Hutton.

These plants were all of small capacity, and those of the C. & N.W. Ry. and of Columbia have been abandoned. The first modern locomotive testing plant capable of handling locomotives of various designs was that of the Pennsylvania Railroad built in 1904. This was designed by the late Axel Vogt, then chief mechanical engineer, assisted by W. F. Kiesel, Jr., the present chief mechanical engineer of the Pennsylvania System. This plant was first installed at St. Louis for the Louisiana Purchase Exposition of 1904, and after operating there throughout the Exposition, was transferred to its present location in Altoona.

The latest plant is that at Illinois University and was built in 1914 under Dr. W. F. M. Goss as Dean of the College of Engineering and Prof. E. W. Schmidt in charge of the Department of Railway Engineering. This plant follows the general design of the plant at Altoona, but provides elaborate arrangements for catching the sparks and cinders thrown out of the stack. For this purpose the exhaust gases are drawn by an exhaust fan through a duct in the roof of the building and are discharged through a centrifugal separator to a 45-ft. stack. The cinders escaping from the locomotive stack are carried by the sweep of the gases through the duct, and those trapped in the separator can be collected and weighed and analyzed.

Details of design of the various plants will be found in various publications listed in a bibliography accompanying the complete paper, of the earlier plants in Dr. Goss's paper presented to the Society in 1904, of the Pennsylvania plant in publications of the Pennsylvania Railroad, and of the Illinois plant in the Engineering Experiment Station Bulletin No. 82.

WORK DONE BY VARIOUS PLANTS

The work done on these plants has been of varying importance. The Chicago & Northwestern Railway plant was of considerable service to the early committees of the A.R.M.M.A. in work on exhaust nozzles, but published no other work. The Columbia University

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Presented at the Regional Meeting, Altoona, Pa., October 5 to 7, 1925, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. Slightly abridged.

plant, situated as it is at Morningside Heights without railroad connections, was adapted only to use the university locomotive for student-instruction purposes. The Illinois University plant has published two reports on tests, one describing complete tests on a single-expansion consolidation locomotive, and the other covering boiler tests on six different sizes of Illinois coal.

Work of Purdue Plant

The widest range of work has been done on the Purdue and the Pennsylvania plants. At Purdue some work of the A.R.M.M.A. committee on locomotive front ends was carried out between November, 1905, and April, 1906, with an Atlantic-type locomotive belonging to the New York Central Railroad. Practically all of the other work has been done with the two experimental locomotives at Purdue University. Both were of the 4–4–0 type. The first, purchased in 1891 as a representative passenger locomotive of that date, weighed about 85,000 lb. in working order. It was replaced in 1897 by a more modern engine weighing about 109,000 lb. with about 61,000 lb. on drivers. Ten years later, in 1907, a Cole superheater was applied to this locomotive; in 1909 the superheating surface was reduced, and in 1911 the locomotive was overhauled and a Schmidt superheater applied. Very many phases of locomotive operation were studied with these locomotives, and the data secured were given freely in papers presented to the Society and to other technical bodies.

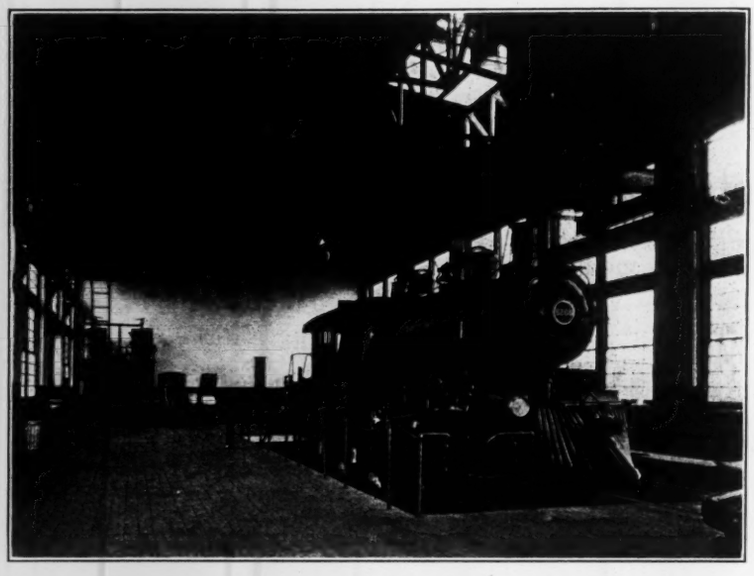

FIG. 1 ATLANTIC-TYPE LOCOMOTIVE 5266, CLASS E2a, SHOWN ON TEST PLANT
(This was the standard P.R.R. passenger locomotive of 1904, and as such has its horsepower curve given in Fig. 4. Tests of this locomotive are reported in P.R.R. Test Plant Bulletin No. 5.)

Writing in 1904 of the information contributed by the Purdue plant, Dr. Goss pointed out that the plant had thrown light on the factors limiting indicated horsepower and rates of combustion which were previously imperfectly understood; that it had shown how the relation between cut-off and steam consumption varied with the speed; and that it had been used to study the action of the exhaust in producing draft and had thus assisted in determining the best proportions for locomotive front ends.

The early work done at the plant consisted in operating the 4–4–0 type locomotive under varying conditions of speed and load. The information obtained was general in character and bore chiefly on variations in tractive force and steam consumption as affected by changes in speed and cut-off. The effect of rate of operation on boiler efficiency was also studied, but as smokebox-gas analyses were not made, no great progress resulted in this direction. A considerable body of definite information was made available, which was undoubtedly of general value to locomotive designers in determining correct sizes for locomotives to perform given service and in securing a proper balance between cylinder power and boiler capacity, but such information was merely supplemental to much of a similar nature that had been secured by road tests, and no basically new information was brought to light. The front-end and stack tests, however, covered research work which could not have been carried out except at a locomotive testing plant. Certain principles previously unknown were brought to light and American locomotive design was definitely affected. It was shown that a tapered stack was preferable to a straight one, and that while a draft pipe would improve the action of a stack which was too small, no combination of draft pipe and stack would give better results than a properly proportioned stack without a draft pipe. It was also shown that that draft-producing action of the exhaust steam was independent of the intermittency of the exhaust, the essential factor in the suction produced being the quantity of steam exhausted. This last fact has been confirmed by all later test-plant results but is often overlooked, particularly when the claim is made that a multi-cylinder locomotive gives a more efficient exhaust action. The results of the various front-end tests were embodied in a series of equations indicating the best proportions for stack and front ends. These equations, while largely empirical, provided a logical basis for the design of the front end and had a considerable influence on American locomotive practice.

The last published information from the Purdue plant dates back fifteen years or more to two reports published respectively in 1907 and 1910 by the Carnegie Foundation. These reports cover two series of tests made by Dr. Goss with the Purdue locomotive, studying, in the earlier series, variations in boiler pressure, and in the latter, variations in superheat. The effect of such variations on boiler and engine operation is shown. In the superheat series fairly accurate heat balances are given, which confirm the information obtained earlier from the Pennsylvania Railroad plant. The two series represent general research work and give definite information on certain points which is of value to the student of locomotive operation.

Work of the Pennsylvania Railroad Test Plant

The installation of the Pennsylvania Railroad locomotive testing plant at the St. Louis Exposition in 1904 opened a new and important era in locomotive testing. The earlier plants had been adapted only for testing light four-coupled locomotives. The Pennsylvania plant was capable of handling locomotives of much greater weight and more varied design. Immediate use was made of the greater plant capacity. The series of tests made at St. Louis covered eight locomotives of widely different designs, and the data secured and published, Locomotive Tests and Exhibits, 1905, were far more complete than any previously available. This test furnished for the first time sufficient information to enable heat balances to be drawn up for a locomotive boiler. Such balances were published by the author in 1908, and gave for the first time exact knowledge as to the relative importance of the various losses which determine the efficiency of the locomotive boiler. Since its transfer to the present location in Altoona the plant has done much work in testing new locomotive designs as produced and in providing a constantly accumulating mass of information which has made possible continued improvements in design. The greater part of the activity of the Altoona plant has been carried out under J. T. Wallis, who was appointed general superintendent of motive power in January, 1912, and who since March, 1920, has been chief of motive power. The direct change of the test plant was in the hands of C. D. Young, engineer of tests, from November, 1911 to May, 1917, and is now in the hands of his successor F. M. Waring.

Results Obtained on the Altoona Test Plant

The very great influence which the work done at the Altoona plant has had on locomotive design is indicated by the following statements as to work done and conclusions reached. These are abstracted from the various Pennsylvania Railroad test-plant bulletins as shown by the reference numbers.

Bulletin No. 9. A self-cleaning front end was developed with diaphragm, exhaust nozzle, and stack arrangement, which gave better results than the form recommended by the Master Mechanics Association.

Bulletin No. 15. After test of an underfeed stoker it was recom-

mended that for each 1900 lb. per hr. of water evaporated there be provided at least one square foot of free air space through the fuel-bed support.

Bulletin No. 21. A comparison was made between results obtained with long and short boiler tubes, the boilers compared being those of the classes K2sa and E6s, having tube dimensions respectively as follows: Length, 21 ft. 0 in. and 13 ft. 8 in.; outside diameter, $2^1/_4$ in. and 2 in.; ratio of length to inside diameter, 125 and 94. When operating with a smokebox vacuum of 6 in. of water the boilers when compared on the basis of evaporation per square foot of heating surface and efficiency gave the following figures:

Long tubes, evaporation 10.8 lb. per sq. ft., boiler efficiency 64 per cent
Short tubes, evaporation 14.7 lb. per sq. ft., boiler efficiency 57 per cent.

A short tube gives free steaming or rapidity of evaporation at the expense of efficiency, so that the most desirable tube length is found by practical considerations as a result of balancing steam production against boiler efficiency. On this basis and taking into account details of construction, the Pennsylvania Railroad adopted as standard for 2-in. tubes a length of 15 ft. or 102 inside diameters, and for $2^1/_4$-in. tubes a length of 19 ft. or 113 inside diameters. The conclusions arrived at were checked against and found to be substantially in accord with those derived from the early experiments made by M. A. Henry on the Paris, Lyons & Mediterranean Railway with much lower rates of evaporation. It was also established that the rate of evaporation is determined by the weight of gases passed over the heating surface and is limited by conditions of combustion and not by a failure of the heating surface to absorb heat.

Bulletin No. 28. The maximum rate of evaporation was established as 7000 lb. of water per hour per square foot of fire area of the tubes.

Bulletin No. 23. It was shown that for piston valves a diameter of 12 in. was sufficient for cylinders up to 27 in. in diameter when using superheated steam, and subsequent tests showed that valves of this diameter could be used with 30-in. cylinders with cut-off limited to half-stroke. Measurements of the stresses in the valve stems gave definite figures showing the great advantage in lightened valve gear to be secured by using valves of the minimum size.

Bulletin No. 24. An extensive series of experiments on superheaters of various designs produced data which are summed up in Fig. 3, showing engine steam consumption for varying conditions of superheat, cut-off, and speed. This diagram gave for the first time, unobscured by other variables, authentic information as to the effect of superheat on steam consumption.

Bulletin No. 27. As a result of earlier tests on the plant certain changes in the design of the E6s type of locomotive were recommended, as follows:

a An exhaust nozzle giving better draft conditions than the plain-circular form
b A screw type of reversing gear
c An enlargement of the cylinders to such a size that the maximum horsepower may be developed at a cut-off not exceeding 30 per cent
d An increase in tube length
e The application of 12-in-diameter piston valves
f A reduction in the width of the center grate bearer.

Tests of the locomotive redesigned as recommended gave a maximum evaporation 15 per cent greater, combined with a boiler efficiency 9 per cent better, than that of the original engine. At the same time the engine performance was generally better, so that a higher drawbar pull was developed at all speeds and the maximum drawbar horsepower was increased over 20 per cent. These advantages were secured with an increase in total weight of only 2.5 per cent. Careful design of the reciprocating parts enabled them to be kept down in weight so that the dynamic augment at 70 m.p.h. was less than 30 per cent of the static load on the drivers.

Bulletin No. 29. Tests were made with a K4s Pacific-type locomotive built in 1914 in accordance with recommendations and experience obtained with locomotives of earlier designs on the testing plant. The coal and water rates and the high thermal efficiency showed this to be the most economical passenger locomotive so far tested on the test plant. For any given amount of fuel fired this locomotive developed more power than any engine previously tested.

Bulletin No. 30. Accurate tests demonstrated the advantage gained in boiler efficiency by the use of a firebrick arch when high-volatile coal is being burned.

Bulletins Nos. 31 and 32. The testing plant having shown definitely the great difference in steam consumption between cut-off at full stroke and cut-off at half stroke, it was logical to attempt to produce a freight locomotive which, even in low-speed drag service, could operate at cut-offs of not more than 50 per cent. The result was the development of the I1s class locomotives with a boiler pressure of 250 lb. per sq. in. instead of 205 and with cylinders enlarged to enable full power to be developed without using a cut-off longer than 50 per cent of the stroke. Compared with the previous standard freight locomotive the new engine with

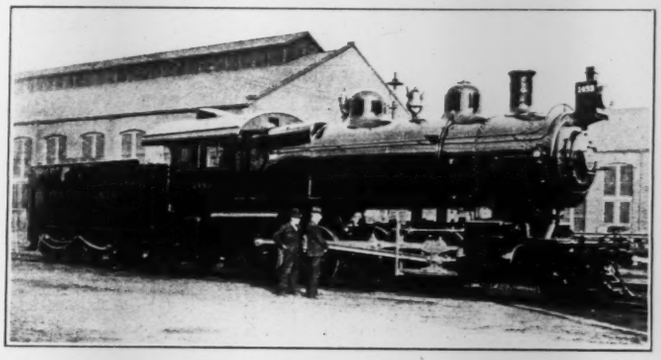

FIG. 2 CONSOLIDATION LOCOMOTIVE 1499, CLASS H2a, THE FIRST LOCOMOTIVE TO BE TESTED ON THE PENNSYLVANIA TEST PLANT AT ST. LOUIS IN 1904

(This locomotive was the standard P.R.R. freight locomotive of that date. The horsepower curve for this engine is given in Fig. 4.)

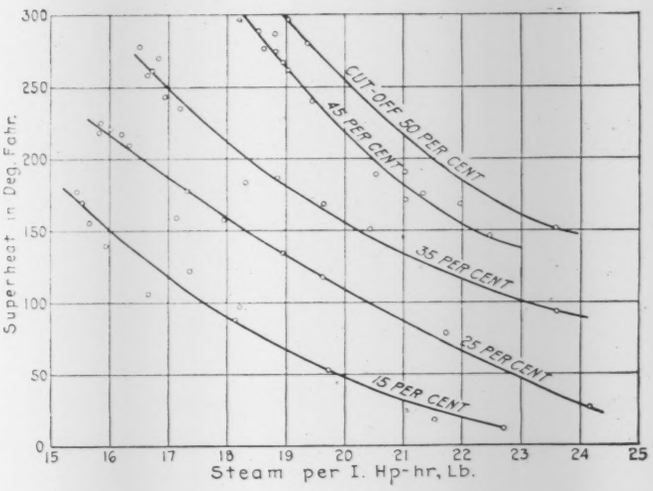

FIG. 3 EFFECT OF SUPERHEAT ON STEAM CONSUMPTION AT VARIOUS SPEEDS AND CUT-OFFS

an increase of 16 per cent in weight gave an increase of 25 per cent in power, and in full gear at low speed showed a reduction of 38 per cent in steam used per indicated horsepower. This type of locomotive with feedwater heater added, giving a further increase in 10 per cent in indicated horsepower, is now the standard freight locomotive of the Pennsylvania System.

NOTES ON RESULTS QUOTED ABOVE

The various statements in the preceding paragraphs which have been quoted in abstract form from the P.R.R. test-plant bulletins show how knowledge has accumulated and how, as a consequence, locomotive designs have been developed and improved. In the majority of cases the tests have been directed to the study of certain definite locomotives, and have been analyzed to determine the possibility of improvements in the design of the engines tested. In a few cases, however, a number of tests have been correlated to permit general conclusions being drawn. Instances of this are the statements as to tube length and as to fire area of tubes. The

latter statement—to the effect that the maximum rate of evaporation was established as 7000 lb. of water per hour per square foot of fire area of tubes—may require some explanation for those who have not studied the test-plant results in detail, and probably would require to be slightly qualified in the light of later test-plant experiments. Flue fire area is given as the limiting factor in evaporative capacity because the tests have shown that evaporation is primarily dependent on the amount of air (gases of combustion) taken through the boiler, and is only secondarily dependent on the area of the heating surface with which the gases of combustion come in contact. If the size and style of flue are unchanged, the relations between fire area, heating surface, evaporation, and boiler efficiency may be summed up as follows:

a An increase in heating surface without an increase in fire area means an increase in tube length. This will give a slightly greater boiler efficiency, but the evaporation will be increased only very slightly, if at all, as owing to the greater resistance which the longer tubes offer to the gases it will not be possible to increase the amount of gas taken through the boiler

b An increase in fire area without an increase in heating surface means an increase in the number of tubes with a proportionate reduction in tube length. Any considerable change in this direction will give a considerable increase in evaporative capacity, because the increased fire area and reduced resistance in length will enable a considerably greater weight of air to pass through under the same draft. At the same time the shortening of the tubes will entail a slight reduction in efficiency

c If a boiler working below its maximum capacity is considered, the fire area may prove to be less important than is indicated above. Take, for example, a boiler fired at the rate of 80 lb. of coal per sq. ft. of grate while the maximum capacity is not reached with less than 120 lb. per sq. ft. of grate area. With an efficient front end it would probably be possible to block up 20 per cent of the flue, and still to draw the same amount of air through the boiler. The fire area would be reduced 20 per cent, but combustion would be at the same rate and the efficiency of heat absorption in the flues would be affected only to the extent of 1 or 2 per cent. The number of tubes being reduced, the amount of gas passing through each tube is increased. The increase in rate of flow increases the rate of heat transfer so that the heat taken up by each flue is increased, and the total amount taken up is practically the same in spite of the reduction in the number of flues acting.

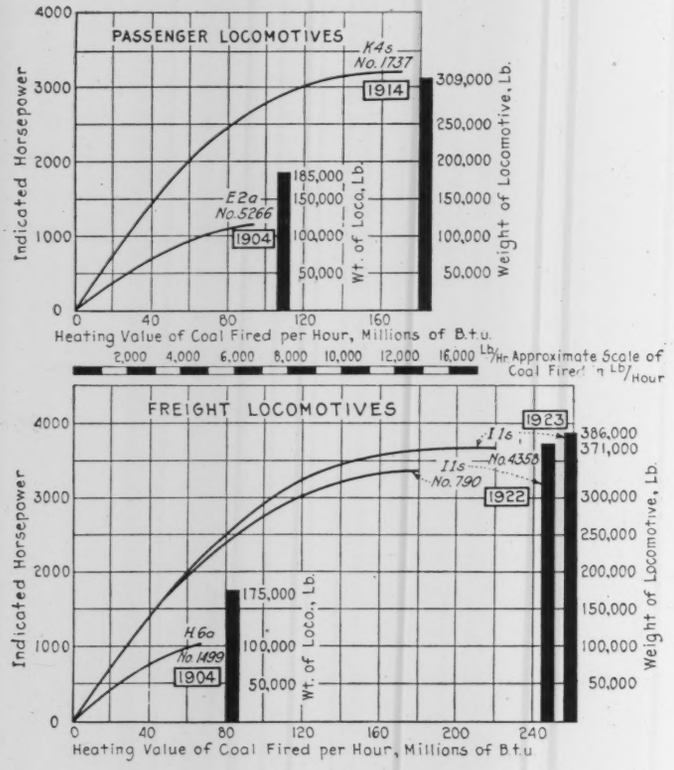

FIG. 4 GROWTH IN SIZE AND EFFICIENCY OF STEAM LOCOMOTIVES

GROWTH IN LOCOMOTIVE SIZE AND EFFICIENCY

To attempt to give a general idea of the growth and development of the P.R.R. locomotives during the time that the locomotive testing plant has been in operation, Fig. 4 has been prepared. It covers both freight and passenger locomotives and compares the engines which were standard in 1904 with those now standard.

TABLE 1 GROWTH IN SIZE AND EFFICIENCY OF LOCOMOTIVES

(Based on maximum rate of operation)

Service	Type	Class	Year built	Loco. weight, lb.	I.hp., max.	Coal fired, lb.	I.hp. per 1000 lb. of loco. wt.	Coal per i.hp. per hr., lb.
Passenger	4–4–2	E2a	1904	185,000	1200	6,000	6.5	5.0
Passenger	4–6–2	K4s	1914	309,000	3300	10,000	10.7	3.0
Increase in per cent							65	
Freight	2–8–0	H6a	1904	175,000	1000	4,500	5.7	4.5
Freight	2–10–0	I1s	1923	386,000	3530	12,800	9.1	3.6
Increase in per cent							60	

The heavy vertical lines are proportional in each case to the weight of locomotive in working order without tender, while the curves give the indicated horsepower in relation to the heating value of the coal fired per hour. The heating value of the coal rather than the weight has been used so that all curves could be plotted on the same base line in spite of slight differences of quality in the coal used. To aid, however, in a general consideration an approximate scale

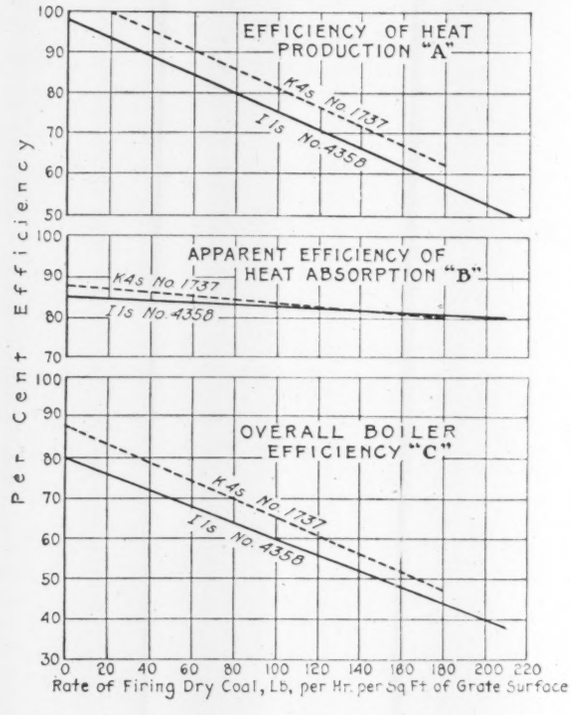
EFFICIENCY OF HEAT PRODUCTION "A"
APPARENT EFFICIENCY OF HEAT ABSORPTION "B"
OVERALL BOILER EFFICIENCY "C"
K4s No. 1737
I1s No. 4358
Per Cent Efficiency
Rate of Firing Dry Coal, Lb. per Hr. per Sq. Ft. of Grate Surface

FIG. 5 LOCOMOTIVE-BOILER EFFICIENCIES IN RELATION TO RATE OF FIRING

of rate of firing in pounds of coal per hour has been added. This is based on a heating value of 14,000 B.t.u. per lb. which is not far from the average value of the coals used. On this basis the figures given in Table 1 for the maximum rate of operation have been obtained.

Attention is called to the horsepower developed per thousand pounds of locomotive weight, which, it will be seen, has been increased by 65 per cent for the passenger and 60 per cent for the freight locomotives.

It is evident that since the Pennsylvania Railroad installed the locomotive testing plant the locomotives have increased not merely in size but also in the efficiency with which the weight and fuel are used. Some of the steps by which the improvement in

efficiency has been obtained have been noted in the abstracts from the various test-plant bulletins.

LOCOMOTIVE-BOILER EFFICIENCY

So far the work of the test plant has been described by abstracts from the bulletins which deal mainly with the solution of concrete problems arising from the testing of certain definite designs of locomotives. The test plant has, however, been of great value in providing material for a study of locomotive operation in general terms, and in thus advancing our knowledge of the subject. Probably the greatest advance has been made in knowledge pertaining to factors which determine boiler efficiency. The state of knowledge before the Pennsylvania plant results had been thoroughly studied is illustrated by an article by the author in 1904 and by H. H. Vaughan's paper before the Canadian Railway Club in 1907. Speculation was active as to the best proportions

Now from the data secured at the locomotive testing plant it is possible to separate the overall boiler efficiency into its component factors. This being done for the boiler efficiencies shown in Fig. 3-C, the lines shown in Figs. 3-A and 3-B represent respectively the efficiencies of combustion and heat absorption as defined above, the method used being explained in an appendix to the complete paper. Numerical values for the three efficiencies at representative rates of firing are given in Table 2.

TABLE 2 LOCOMOTIVE-BOILER EFFICIENCIES

Locomotive	Rate of firing, lb. per sq. ft. grate per hr.	Efficiency of heat production, per cent	Efficiency of heat absorption, per cent	Overall boiler efficiency per cent
K4s	80	86.0	85.5	70.0
	120	76.5	83.5	61.0
	160	67.0	81.5	52.0
H1s	80	79.7	84.4	64.0
	120	70.6	83.2	56.0
	160	61.5	82.0	48.0

FIG. 6 ATLANTIC-TYPE LOCOMOTIVE 89, CLASS E6s, BUILT IN 1912

(Represents the standard P.R.R. Atlantic-type locomotive with the addition of a superheater. Tests of this locomotive are reported in Bulletin No. 27. By comparing the boiler tests of this engine with those of the K2s Pacific type in Bulletin No. 18, information as to the relative value of long and short boiler tubes was obtained.)

to be assigned to firebox and heating surface, but very little was definitely known. Dr. Goss's work had indicated that the losses by incomplete combustion were large, but with no information as to the amount of air consumed, accurate information as to the details of combustion and heat absorption was lacking. Against this set the fact that during the recent tests of the Missouri Pacific three-cylinder locomotive the author was shown a predicted curve of smokebox temperatures which in the majority of cases did not show differences of more than 15 deg. fahr. from the results actually obtained.

As a concrete illustration of the information which can be derived from the test-plant data by further analysis, there is given below a brief account of some work which the author has undertaken recently in connection with a study of locomotive-boiler efficiency.

In all tests of locomotive boilers in which the boiler efficiency is measured over a considerable range of rates of firing it has been found that as the rate of firing increases there is a drop in boiler efficiency which can be represented best by a straight-line law. In Fig. 5-C the lines found for the Pennsylvania Railroad K4s passenger and the H1s freight locomotive are shown. The boiler efficiency here is the heat taken up by the boiler in evaporation and superheating expressed as a percentage of the heating value of the coal fired. The value found for this efficiency at any given rate of firing is dependent on three factors:

1 The efficiency of combustion, which is the efficiency with which the coal is burned in the firebox. In other words, the amount of heat actually produced in the firebox expressed as a percentage of the heating value of the coal fired

2 The efficiency with which the heat produced is taken up by the boiler

3 The amount of heat taken up by the boiler which is lost by external radiation. In the absence of exact information the author assumes this external loss to be 5 per cent of the heat utilized in evaporation. This assumption has been justified by the results obtained.

These figures are not offered here for the purpose of making a detailed comparison of the K4s and the H1s locomotives, but are given as representatives of the results obtained in good locomotive practice. It is characteristic of the locomotive boiler to show efficiency relations generally similar to those in the tables. The efficiency of heat absorption shows high values which remain nearly constant, being little affected by the rate of firing. The efficiency of combustion shows a large variation with the rate of firing. At low rates the efficiency is very close to 100 per cent, but falls rapidly as the rate of firing is increased. At a normal maximum rate of 100 to 120 lb. per square foot of grate per hour the efficiency of combustion is in the neighborhood of 60 per cent, while if the boilers are pushed to actual maximum power the efficiency falls to less than 50 per cent. The efficiency of heat absorption is, as has been said, much more uniform, never dropping below 80 per cent at the highest rate of firing and showing values of about 86 per cent at low rates of firing.

Now it is evident that if the overall efficiency of the boiler is to be improved it is desirable to study closely the two component efficiencies. In the first place, it is noted that at normal rates of firing the efficiency of combustion is in the neighborhood of 60 per cent, and the efficiency of heat absorption is over 80 per cent. This indicates that there is a greater margin for improvement in combustion than in absorption. In reality, the possible margin for improvement in absorption is even less than would appear from these figures, for while it is possible to have 100 per cent efficiency of combustion with low rates of firing, it is not physically possible for the efficiency of heat absorption, as defined above, to have a higher value than about 91 per cent. This follows because it is physically impossible, even under ideal conditions, for the boiler to take up all of the heat produced in the firebox. Under no conditions is it possible to cool the smokebox gases below the temperature of the saturated steam in the boiler. With a boiler pressure of 205 lb. per sq. in. this gives an ideal minimum temperature of 390 deg. fahr. for the smokebox gases, at which temperature each pound of gas will carry 96 B.t.u. Under normal conditions the production of one pound of combustion gases will

be accompanied by the liberation of approximately 1080 B.t.u., and with an atmospheric temperature of 65 deg. fahr. adding a further 13 B.t.u., the total heat offered to the heating surface will be approximately 1093 B.t.u. per pound of gases of combustion. With ideal conditions by which the gases of combustion were cooled to the saturated-steam temperature, 96 B.t.u. would be left in, and 997 B.t.u. absorbed from, each pound of gas. That is, under ideal conditions the heating surface cannot possibly take up more than 91 per cent of the original heat in the gases. Let us designate as "apparent efficiency" the efficiency of heat absorption based on heat produced, and designate as "real efficiency of absorption" the heat taken up expressed as a percentage of the heat absorbable under ideal conditions. Then while the figures above show "apparent" efficiencies of heat absorption ranging from 80 to 86 per cent, the "real" efficiencies range from 94 per cent at low to 88 per cent at high rates of firing. It is evident that as a piece of heat-interchange apparatus the modern locomotive boiler, as exemplified by the Pennsylvania Railroad boilers in question, is highly efficient and that there is but a small margin for improvement in heat absorption.

The efficiency of combustion shows a different condition. Here an efficiency of 100 per cent is possible and is shown by the tests to be reached at low rates of combustion. At the usual rates of operation, however, the values are generally in the neighborhood of 60 per cent or less.

There is evidently considerable field for improvement here, but it must not be assumed that improvement is easy. There is yet room for considerable study before the final answer is given. The diagrams show that boiler efficiency can be improved by reducing the rate of firing per square foot of grate. If the same power is to be maintained, this means an increase in the size of grate so that the total amount of coal burned may be maintained. Locomotive designs adopted by various railroads in the last few months show development along this line, but as yet no testing-plant analysis of performance is available.

In this connection the author calls attention to the fact that in the discussion of boiler efficiency in relation to rate of operation it is usual, as has been done above, to measure the rate of firing in terms of coal per square foot of grate per hour. This assumes that the grate area is the controlling factor in determining the efficiency of combustion. Grate area is an important factor, but not the only one. Firebox volume must also be considered. The test-plant data are not yet sufficiently complete to permit a determination of the relative values of grate area and firebox volume, but one series of tests with high-volatile coal suggests that with this fuel the volume is the more important. The point deserves more study and it may be found that combustion efficiency can be best improved by a slight increase in grate and a considerable increase in firebox volume with appropriate arrangement for giving a long flameway.

CONCLUSION

For the future we look with confidence for a continued growth in the use of the locomotive testing plant. The great increase in locomotive efficiency which has been seen to have taken place in the last twenty years makes further advance more difficult, and makes exact knowledge a necessary condition for such advance. As it is the function of the locomotive testing plant to provide exact knowledge, it is not surprising that new plants are under consideration by railroads and by locomotive builders. A very great step forward could be made if the American Railway Association were to construct or to take over a locomotive testing plant to be devoted to the scientific and impartial study of locomotive designs and devices, and to undertake research work concerned with the basic scientific laws governing locomotive operation. Correct locomotive design is based on definite natural laws, knowledge of which can be obtained only by accurate experiment. The great influence which the locomotive testing plant has had on locomotive design is due to the opportunity it has given for thorough and accurate study of the fundamental laws governing locomotive operation. It is evident that the wider and the more accurate the knowledge a designer has regarding the laws governing the operation of his product, the more confidently and the more successfully can he proceed with improvements in design.

The Fundamentals of Research

BY WM. L. DEBAUFRE,[1] LINCOLN, NEB.

RESEARCH in engineering, or applied science, must be carried to fruition by the installation of a new industrial process or machine or the production of a new article of manufacture or an improvement thereon, or it must show that a proposed improvement is practically impossible. In the past, new processes and new materials were often the result of accidents. Today, however, they are the results of engineering research conducted according to the scientific method, which consists in the following fourteen steps:

1 A statement of the object of the research project to develop a certain new process, machine, article of manufacture, or improvement thereon, and of the plans for accomplishing this object.
2 A search to determine what has been done along similar lines in the past.
3 A summary of the typical characteristics of old processes, machines, or articles of manufacture.
4 A theoretical analysis of such typical characteristics to discover the relations involved.
5 A determination of the numerical constants required in these relations.
6 The deduction of a new process, machine, or article of manufacture based on the theoretical relations discovered.
7 Laboratory experiments to determine the truth of theoretical deductions.
8 The design of an apparatus of sufficient size to illustrate the performance of a full-scale unit.
9 The building and testing of such a semi-commercial unit to determine its characteristics.
10 The theoretical analysis of the test data for the semi-commercial unit.
11 The design of a full-size unit.
12 The building and testing of a full-size unit.
13 The theoretical analysis of the test data for the full-size unit.
14 The preparation of a report upon the whole development.

In experimental work much planning is required for a comparatively small amount of work, yet it cannot be curtailed with safety, for such curtailing might render the experimental work useless. Of even greater importance is a search to determine whether the whole problem has been solved previously. The following aids are available for a library search:

a The Engineering Index, published by The American Society of Mechanical Engineers.
b The Industrial Arts Index, published by the H. W. Wilson Co., New York.
c Science Abstracts, Section A—Physics, Section B—Electrical Engineering, published by The Institution of Electrical Engineers and The Physical Society of London, England, in coöperation with the American Institute of Electrical Engineers and the American Physical Society.
d Chemical Abstracts, published by The American Chemical Society.
e Journal of the Chemical Society of London, Abstracts, published by the Society.
f International Catalogue of Scientific Literature, published by the Royal Society of London.
g Glazebrook's Dictionary of Applied Physics, 1923.
h Thorpe's Dictionary of Applied Chemistry, 1921.
i The Official Gazette of the U. S. Patent Office, published by the U. S. Government Printing Office.

Mention should also be made of the Library Service Bureau of the national engineering societies in New York City. The National Research Council also maintains a Research Information Service. The publications of various Government Bureaus and of the Engineering Experiment Stations connected with the state and privately endowed educational institutions should not be passed over without notice as a source of much valuable information.

The files of the U. S. Patent Office should be searched, not only for light on previous work done along the same lines but also to learn whether patent protection is possible should there be developed an improved process, machine, or article of manufacture.

As the value of the time and effort expanded upon the research is in general dependent upon the knowledge acquired by others of the results, a report thereon is a fundamental requirement. The methods of calibrating instruments and the details of conducting the test should be minutely described in the report.

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From a paper presented at the Annual Meeting of the American Society of Agricultural Engineers, Lincoln, Neb., June 20, 1924.

Lubricants and Lubrication

The Supply of Lubricating Oils—Theory of Lubrication—Cylinder Oils—Lubrication of Cylindrical Bearings—Bearing Loads and Temperatures—Application of Lubricants to Bearings—Power Losses—Packing Waste—Reclamation of Used Packing and Oil—Cost of Lubrication

By JAMES DUGUID,[1] FRANKLIN, PA.

PREVIOUS to 1859 commercial lubricants were derived mostly from vegetable and animal oils, and from a small amount of mineral oil obtained from the distillation of shale. However, as far back as 1797 crude mineral petroleum was found and skimmed from the waters of Oil Creek, near Titusville, Pa., and is reputed to have sold for as high as $16 per gallon. But the real production of mineral oils began when in 1859 Col. Edwin L. Drake drilled the first oil well on the banks of Oil Creek. In that year 1873 barrels of crude petroleum were produced and sold at an average price of $20 per barrel. A monument in memory of Colonel Drake—the real pioneer of the oil industry—has been erected and now stands near the locality of his discovery well.

At the close of the year 1924 the average monthly production of crude oil for the year had been about 59,000,000 barrels or nearly 2½ million gallons per month, whereas in 1918 the total production was about 30,000,000 barrels per month and less than half our present average production. This will give some idea of the very largely increased production developed in the past few years.

The total crude mineral petroleum produced since 1859 in the United States is a little less than eight billion barrels of 42 gallons each. Various prominent geologists have estimated our total oil resources at from twelve to fourteen billion barrels. More than 600,000 wells have been drilled in the United States and the average cost per well has been estimated at about $20,000, which is probably a little high for a general average. In 1923 more than 24,000 wells were drilled and in 1924, nearly 22,000 wells, but about 25 per cent of the latter turned out to be dry holes. The percentage of dry holes is increasing each year, which tends to bear out the geologists' estimates of diminishing supply.

The prices of the various crudes range from about 50 cents to $3.30 a barrel, the latter being the posted price for Pennsylvania crude at the present time. It is not uncommon to have considerable premiums above the market price offered for this grade of oil.

Of the total crude produced in the United States during the year 1924, about 4 per cent, or more than one billion gallons, was converted into lubricants of various kinds. Of this amount approximately one-third was exported, the remaining two-thirds going to domestic consumption and stocks. The amount in storage today is somewhat over 250 million gallons. The foregoing figures give a general idea of the developments of the industry and its importance at the present time.

Early Work in Lubrication

Down to the year 1875 the question of lubrication was not given much attention from a scientific engineering standpoint. However, about 50 years previous to that time it had become the subject of much speculation and discussion, and after considerable experimental research Morin advanced certain theories on which he established the following three laws of friction:

1. The friction between two bodies is directly proportional to the pressure.
2. The amount of friction (pressure being the same) is independent of the areas in contact.
3. The friction is independent of the velocity, although static friction is greater than the friction of motion.

About 1875, however, these laws or theories were found to be almost entirely erroneous and it was proven that the friction between two bodies is *not* directly proportional to the pressure; that the amount of friction *is* dependent upon the area in contact, and that the friction is also dependent upon the velocity, but that in neither case are they in direct proportion. Notwithstanding all the technical data that have become available since that time on friction and lubrication, it is evident that we are to a degree still groping in the dark regarding these engineering subjects. The various kinds of metals used for bearings, their mechanical construction, the various kinds of lubricants and the methods of applying them—all of these as related to bearings used for the same purpose show a lack of engineering standards as well as of definite knowledge of the subject.

Even at the present time, with all the modern developments of engineering science, very few users of lubricants employ engineers whose duty it is to look into the question of the efficiency of the lubricants they buy, the cheapest oils usually being favored without regard to the total amount used or to the losses that occur through high friction and wear.

It is only common-sense that large users of lubricants should require the oil manufacturer to demonstrate the efficiency of the lubricants he supplies. If this were always done it would soon become apparent that price, appearance, and viscosity of oils provide a very unreliable basis for selection. The factor of viscosity is mentioned particularly because this can be so easily faked. Very light oils of no particular lubricating value or viscosity can be brought up to a very high viscosity and fine appearance by compounding them with either a small proportion of degras oil or aluminum soap, and such oils, when placed in cans bearing fancy colored and extravagantly worded labels, are very attractive to the uninitiated.

It is evident that at the present time the efficiency of lubricants—as well as the efficiency of other materials—is being more studied than in the past. This has been brought about mainly through increased costs of labor and material, and the modern manufacturer must make close inquiry into actual efficiencies in order to maintain his works in profitable operation.

A few years ago railroads generally did not give much attention to the efficiency of the individual locomotive—they kept up to the requirements of tractive power by building larger ones. But it is quite apparent that the locomotive has now about reached its limit for size, and the live question is to make each unit more efficient. One of the results of this is that we now find locomotives equipped with feedwater heaters—who would have ventured suggesting such cumbersome apparatus to the railroad official a few years ago?—superheaters, and many other auxiliary devices. In fact, progressive railroad men are now willing to try out any device that gives reasonable promise of even a small percentage of increase in locomotive efficiency. In the author's opinion, the next important problem that will receive close attention is the friction of the lubricated parts, as here is surely a fertile field for investigation and improvement.

The efficiency of a machine is measured by dividing the amount of useful work performed by the gross work of the machine. Experience has shown that friction is the principal cause of loss of energy, and therefore loss of work in machinery; also that waste energy can be reduced to a minimum by installing the proper kind of rubbing surfaces and by the use of the best lubricants.

The object of lubrication is both to reduce friction and to prevent excessive temperatures. The true value of the lubricant depends upon its efficiency in reducing friction, its durability under wear and adverse mechanical conditions, its freedom from acids and grit, from liability to gum, and its physical condition when subjected to change in temperature. It should always be borne in mind that as the temperature increases the lubricating power of the lubricant decreases, and it should not be assumed that

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Presented during Gas and Oil Power Week at a joint meeting of local sections of the A.S.M.E., A.C.S., and S.A.E., the Engineers' Club of Lincoln, and the A.S.M.E. Student Branch, University of Nebraska, Lincoln, Neb., April 25, 1925. Slightly abridged.

a hot bearing may be cooled by increasing the quantity of the lubricant. The cause of the heating must be removed.

Tests that usually interest oil users are those covering specific gravity and viscosity, but from a practical lubricating standpoint these tests as ordinarily carried on in commercial practice count for little, other than for classifying the oils. They do not furnish any absolute data for determining the real lubricating value of the oil, although enabling certain conclusions (sometimes wrong) to be drawn on the subject. The viscosity of lubricating oils is ordinarily taken at 100 or 130 deg. fahr., and it is therefore not known what the viscosity of the oil is when the film is passing under the highest point of load on the bearing. There are important factors or characteristics in various lubricants that greatly affect their lubricating value which cannot be determined by ordinary gravity or viscosity tests unless the latter are carried out under extreme pressures and temperatures such as are not practical for commercial testing. One important feature called by some authorities "oiliness of the lubricant," cannot be ascertained without special research tests.

THE THEORY OF LUBRICATION

The theory of lubrication is altogether too technical to be set forth in a paper of this kind, and therefore only a few of the propositions that are readily understood and perhaps of the most interest to the ordinary oil user will be referred to. Some of these are roughly as follows: That irrespective of how smooth a journal and bearing may appear and no matter the amount of so-called finish that may be applied to them, there are still projections presented on these apparently smooth surfaces and these projections on the journal and bearing will interlock and abrade when set in motion if not separated by a lubricating film. . . That one oil film adheres to the bearing and one to the journal, and that there is no slipping of these films on the metals. . . That the films slide on each other and the metals are kept wholly separated, and therefore with this condition there can be no wear of the metal parts. . . That the oil film is in the shape of an eccentric ring, the thinnest section being a few degrees past the center of the bearing load in the direction of motion of the journal. . . That the pressure of the oil film varies, being greatest (about twice the pressure per square inch as compared with the load on the bearing) at a point a few degrees past the center of the bearing load, and that the sum of the pressures on the oil film about equals the total load on the bearing. . . That the temperature of the oil film passing under the area of highest pressure is not known. . . That the resistance of one oil film to sliding on the other is the frictional load, and that the heat generated by such resistance causes the temperature. . . That the thickness of the oil film under the loadarea varies from about 0.0002 in. to 0.003 in. in thickness, depending on the class of bearing, velocity, elastic deformation, etc.

In bearing lubrication it is evident that in a majority of cases the lubricating film is very thin, and that with the ordinary mechanical methods of applying the lubricant there is difficulty in feeding the oil to the bearing so as to maintain a uniform film. It is also evident that the performance or condition of the oil film under the maximum pressures of load are still in the research stage, and that infinitesimal differences not observable by the naked eye on the surface of journals and bearings of the same class affect the oil film and make radical differences in running temperatures.

It has been demonstrated that when a good oil film is established it does not wear out rapidly and will last for an almost incredible period of time, provided its frictional temperature is normal. If the temperature is increased above normal either by friction or some foreign heating agent, the life of the film will be very materially shortened and its duration will have a closer relation to temperature than to the lubricated surface rubbed over.

KINDS OF LUBRICANTS

Practically all commercial lubricating oils are compounded; that is, they are mixed with other lubricants for the purpose of obtaining the viscosity of body, oiliness, and emulsifying or non-emulsifying properties that will make each lubricant best suited for the purpose it is to be used.

It is sometimes assumed by oil users that none but mineral oils are needed in the compounding of oils used in the presence of steam. This is not the case, as both vegetable and animal oils such as castor, rape, tallow, lard, neatsfoot, whale, wool oil, and fish oil are all used in the compounding of various kinds of lubricating oils. In the greases there are also used such solid lubricants as graphite, talc, and mica, while leads are used both in oils and greases. In general, lubricating oils may be divided into seven classes: namely, unfiltered cylinder oils, filtered cylinder oils, red oils, pale oils, neutral oils, white oils, and black oils.

WHERE LUBRICANTS ARE USED

While there are no complete or accurate data showing in detail the consumption of lubricating oils, it must be quite evident that a very considerable percentage of it is wasted by careless handling, excess and faulty application, and improper mechanical conditions. It seems that few users realize that it requires only a comparatively small amount of suitable oil to lubricate a rubbing surface. The average user seems to work on the theory "that if one pint of oil a day will lubricate a machine efficiently, then two pints per day will lubricate it twice as efficiently." Or, judging from observation, some seem to think that if oil is plastered all over the machine it has then—but not until then—the proper appearance of efficient oiling. These really seem to be the standard gages for application in only too many cases.

Railroads, generally, are without question the most efficient handlers of lubricants and are at the present time lubricating their locomotives for about $4.50 per 1000 miles and passenger and freight cars for about 4 cents per 1000 miles, with an average of about 6 cents per 1000 miles for passenger cars. On railroad steam locomotives the average lubricated surface rubbed over per mile for all types of equipment is approximately 80,000 sq. ft., and the average consumption of lubricants is at the rate of about 90 miles to the gallon; which makes, roughly, 7,200,000 sq. ft. of surface rubbed over per gallon, or nearly 2 sq. ft. per milligram of lubricant.

Railroad freight cars make an average mileage of about 30 miles per day and the amount of lubricant used is about $1^1/_4$ pints per 1000 miles. This is equivalent to 680,000,000 sq. in. of lubricated surface rubbed over per pint of oil, or one milligram of oil per 1500 sq. in. Passenger cars use an average of 2 pints of oil per 1000 miles, but on account of the larger and greater number of journals, their performance about equals that of freight cars.

There are now, in this country alone, more than 15 million automobiles and about $1^1/_2$ million trucks. The average consumption of gasoline for autos is about 335 gal. per year and for trucks 590 gal.—altogether about 420,000,000 gal. per month. The consumption of lubricating oils for these machines is estimated to be around 4,200,000 gal. per month.

These figures give some idea of our domestic requirements for lubricating oils, although there are so many other established uses for lubricants that it would be an almost endless task to enumerate them.

APPLICATION OF LUBRICANTS TO BEARINGS

Touching briefly on the various methods of applying lubricants to bearings and journals, the most generally used are the hand-feed system, where oil is applied through an oil hole in the bearing; and the oil-cup system, where the oil feed is regulated by a siphon wick, or through a limited opening regulated by an adjusting valve. The circulating system may operate on the gravity method, with pumps to return the oil to the gravity tanks, or on the power pressure system, where pressure up to 1000 lb. per sq. in. is sometimes used, as for turbine thrust bearings.

Ring oilers are used on machines such as electric generators and motors, oil pads on steam and electric cars, and waste packing for journal lubrication. Splash systems are used on many internal-combustion engines and on some types of steam engines. There are also nearly as many varied methods of applying grease lubrication.

The different mechanical conditions under which lubricants must be applied make the use of all of these methods necessary; but where the circulating system can be installed it is undoubtedly the best method of application, for the reason that a steady, uniform quantity of lubricant can be fed to the bearing, which oil, after it has done its work, passes on to the precipitating filter, and

carrying with it some of the heat generated in the bearing, passes through cloth filters which take out the small floating particles, and is again circulated to the bearing in a cool condition.

The capacities of the filters required for stationary steam engines are approximately as follows:

	Gal. per hour	
Horsepower	Slow speed	High speed
100	4	5.5
500	12	15
1000	21	..
2000	75	..

These liberal volumes provide ample time for the circulating lubricant to reduce to normal temperature; also to separate in the precipitating filter.

As the company with which the author is connected—specializing exclusively in the manufacture of steam- and electric-railroad oils—alone manufactures more than 200 different kinds of lubricants, it would be an unnecessary and practically endless task to attempt to describe, or even name in detail, the various different lubricants or outline their uses, method of application, or their performance in service. Therefore, only a few of the kinds in most common use will here be considered.

Cylinder Oils

Examining average specifications for cylinder oils used for both low and high steam pressures, it is found that the following physical characteristics are demanded:

For low steam pressures (where high-grade oil is not required) the oil—
- Must contain not less than 5 per cent acidless animal oil
- Must have a flash test of at least 450 deg. fahr.
- Must have a fire test of at least 500 deg. fahr.
- Must have a cold test of at least below 50 deg. fahr.
- Viscosity at 212 deg. must not be less than 100 sec.

For high steam pressures the oil—
- Must contain not less than 5 per cent acidless animal oil
- Must have a flash test of at least 550 deg. fahr.
- Must have a fire test of at least 575 deg. fahr.
- Viscosity at 212 deg. must not be less than 150 sec.

Cylinder and valve lubrication—especially on steam locomotives—present perhaps the greatest of lubricating difficulties, while high-pressure stationary power is comparatively free from difficulty, owing to the fact that high steam temperatures will readily atomize a good-grade oil and that there is a minimum amount of saturation in the cylinders. On low pressure there is a lack of the higher temperature necessary to atomize a good oil, and the condensate in the cylinders is excessive.

The difficulty of locomotive-cylinder lubrication is principally due to the fact that the exhaust pipe terminates in the smokebox and, when drifting, the vacuums in the cylinders—at the time the exhaust valve opens—cause a rush of coal gases into the cylinders which, as may be readily understood, is most destructive to lubrication. Again, the locomotive's continuous starting and stopping cause a variety of saturated conditions in the cylinders.

In applying cylinder oils the unit of volume of feed is a certain number of drops per minute, depending on the size and other conditions of the engine. The volume of these drops from various types of lubricators are as follows:

Diam. of drop, in.	Volume of drop, cu. in.	No. of drops per pint
1/8	0.00102	28240
3/16	0.00345	8369
1/4	0.00818	5530
5/16	0.01598	1807

In an atmosphere of saturated steam the heavy cylinder oil before mentioned will have about the following percentage of volatilization:

Temperature, deg. fahr.	400	500	600	700	800
Percentage of volatilization	0.5	5	18.9	44	99

The remainder of the oil (if an excess amount has not been applied) will atomize in the steam and be carried to and gather on the surfaces of the cylinder and valves. It is possible to lubricate valves and cylinders using steam at 600 deg. fahr., with a good lubricant of 525 deg. fahr. flash point, and this may be done successfully with the right quality of valve oil, although it is sometimes difficult to convince those in charge of engines using high-pressure superheated steam that it can be, as they insist that the flash point should be higher than the maximum temperature of the steam. This idea was mostly prevalent at the time of the advent of superheated-steam locomotives, and cylinder oils were accordingly made of a very high flash test. But the results were so unsatisfactory due to the fact that such oil did not volatilize or atomize readily in the steam, that the mistake of this theory was evident.

Exhaustive tests have shown that a high-grade cylinder oil with a flash test of 525 deg. fahr. will not burn in a cylinder even at a steam temperature of 1000 deg. fahr., if gases and atmosphere are excluded. The oil can be discerned under these conditions in the form of mist or fog, but if air or other gases are admitted at this period the oil will immediately burn. The critical temperature is apparently above 560 deg., which shows that locomotives must not be shut off entirely when the cylinder temperature is at a maximum.

Another trouble encountered in cylinders and valves is in the form of deposits commonly but erroneously called carbon. But if a good grade of oil has been used and these deposits are analyzed, it will be found that there is but a small trace of carbon, and the average analysis will show that on stationary engines the deposits consist principally of magnesium, sodium, etc. which have been carried over from the boilers, and which, when combined with the oil in the cylinder together with a little iron rust, form a deposit that is often blamed on the oil. A common form of deposit, when analyzed, is shown to consist of magnesium, sodium, or mud, 52 per cent; petroleum oil with an amount of saponifiable matter, 30 per cent; oxide of iron, 15 per cent; carbon, 3 per cent; or in the case of cylinders of locomotives using superheated steam, carbonaceous combustible, 59 per cent; thick baked oil, 13 per cent; oily, gummy matter, 3 per cent; red ash (iron oxide), 25 per cent; small amount of carbon.

A well-lubricated cylinder should, on the removal of the cylinder head, appear dark and smooth; a light rubbing with a piece of paper should produce a bright polished surface. If the cylinder walls appear bright on first inspection, it is a proof of inefficient lubrication and of high frictional wear. If the cylinder deposits show a high percentage of red oxide of iron (this should not be confused with iron rust) it shows there is an excess of free fatty acid in the lubricant. If the cylinder shows lack of lubrication on the bottom, or in spots elsewhere, it is evident that the oil is lacking in a proper percentage of animal oil.

The presence of carbon and high percentages of carbonaceous combustible in the cylinders of locomotives may be construed as evidence of poor handling on the part of the engineman. If the cylinders show undue wear on any particular side, this is evidence of faulty lining of the crosshead slides. If oil is found in the steam chambers of the valves and there is an apparent lack of lubrication on the walls, it shows that the lubricant is too heavy in body to be atomized at the prevailing steam temperature. If oil is found in these chambers or there is an unusually heavy coating on the walls, an excessive quantity of oil is being applied.

The actual load caused by the frictional resistance of the valves and cylinders is not great, but the losses in power and steam due to the wear of these parts is a very important consideration; such wear causes loss of steam to the exhaust and faulty steam distribution, and when it is remembered that even with 100 per cent perfect mechanical conditions only a small percentage of the heat of the coal fired is available as power at the piston rod, then any undue or abnormal wear of these parts causes serious loss. Therefore it is plain that a thorough examination of steam cylinders that are giving trouble should disclose evidence that would tend to locate the trouble.

Lubrication of Cylindrical Bearings

In most of the printed instructions on the lubrication of cylindrical bearings will be found the following injunction: "Use an oil of just sufficient viscosity so that the film will bear the load at the required pressure per square inch and at the required journal velocity."

To the author's mind such instructions are much the same as advising the building of a machine to stand a load of 100 lb., with-

out providing any factor of safety to insure that it will not fail at a load of 101 lb. A proper—even though a conservative—safety factor is just as necessary for lubricants as for any power mechanism, because the lubricants are subjected to just as many unusual loads and strains. This makes it necessary to pay some penalty in friction of the lubricant as a factor of safety, just as one pays for excess weight in a machine for the same purpose; for if such provision is not made, constant failures will result.

Conditions can readily be pictured where the safety factor of the lubricant requires a comparatively small margin, such as in power plants where speeds are uniform, where the foundations of all bearings are rigid, where a circulating oil system is used, and also where—it goes without saying—supervision and repairs are efficient. But a very small percentage of lubricants is used under such conditions.

An absolutely ideal lubrication performance is not possible, or at least is not practicable, to obtain. This for the reason that it would require an almost endless number of different oils to lubricate even an ordinary machine, on account of the widely varying velocities, loads, and conditions under which the machine operates. If the general theory of lubrication is correct, as we are all inclined to agree, then any one kind of a lubricant has but a restricted range for efficient lubrication, and the lubricant which does not keep solid friction and wear at a minimum under all conditions does not fully meet the requirements of a lubricant in the fullest sense.

A condition met with in lubrication that does not seem to be fully apparent to the average user of oils, is that frequently there are chemical or physical imperfections in the metals of the bearings or journals, or some mechanical defects in the fitting of these parts that are not discernable by the naked eye. Yet tests will clearly indicate that excess friction and correspondingly high temperatures can be, and often are, caused by variations in the smoothness of the bearing or journal surfaces that cannot be easily detected, or in fact cannot be detected with the naked eye.

The author has at various times carried out temperature and friction tests on several sets of bearings and journals made from the same materials and with the same mechanical construction and adjustment of the bearing fit as closely as it was possible to make them with micrometers, but when tested with the same lubricants, loads, velocities, and atmospheric temperatures they produced entirely different frictions and temperatures, and a thorough calibration and examination did not reveal any mechanical difference in the bearings. The inference should not be drawn from this that either of these bearings would give trouble by excess heating in service, but the example goes to prove that even infinitesimal defects may cause heating, and that in tests it is a difficult matter to determine fully or accurately the true friction of any lubricant.

As a practical illustration of this: At one of the largest and most important passenger terminals on a certain large railroad, where about 3000 pairs of wheels were changed per year, there was a general passenger-car foreman who was familiarly known as the "journal bug," due to the fact that he made a particular hobby of the surface finish of journals. After a journal was removed from the turning lathe with an apparently perfect finish, he would have it placed in a pair of improved centers and run a lead lap lubricated with oil on it until another journal was turned; these journals were then thoroughly protected until placed in a car. What was the result? Although the majority of cars were of the heavy steel type and made the initial run of 142 miles without a stop, during the year considered only two journals out of the six thousand applied gave any serious trouble by heating on the initial run. This resulted from finishing a journal beyond what would ordinarily be called perfect. Now, if the oil film used in lubrication is of the thickness claimed by some authorities, why should finishing journals beyond an apparently good face thus improve their initial running temperatures?

Hot Bearings

On heavy machinery, and particularly on railroad equipment, heated bearings are not only of common occurrence but are considered a costly and dangerous menace to efficient and economical operation.

What is the cause of a bearing "running hot," as it is commonly termed? There is just one cause and that is the absence of the oil film between the bearing and the journal on the area of a bearing load. This oil film may have been eliminated through any one or more of various causes, such as lack of oil supply, application of the oil at a wrong point with relation to load point on the bearing, oil of too low viscosity at running temperatures to carry the bearing load, journal velocity either too low or too high in relation to oil viscosity, bearing fitted too tight to journal, load on the bearing unequally distributed, packing not in contact with journal, packing glazed on surface next to the journal due to lack of oil on such surface, new brasses applied without a sealed crown fit—allowing oil to by pass through openings in crown, oil filtering down to the bottom of the oil box where it cannot be brought up to the bearing by capillary attraction due to packing-waste condition and location. The author has never yet seen a hot box in railroad service due to dearth of oil in the box, but certainly many due to lack of oil on the journal.

Now, just what takes place in a properly fitted journal and bearing when it starts to heat or when it has become heated to a degree that will cause abrasion and fusing of the metals, assuming that the oil used is too low in viscosity to maintain the film when excess speed or load is applied? Simply this: the oil gradually reduces in viscosity until it has not sufficient body to carry the load and then solid friction (which is about ten times that of liquid or oil friction) takes place and there is a correspondingly rapid increase in temperature that cannot be radiated by the surrounding mass. This condition is further aggravated at this point by another: the heat generated in the journal is radiated in both directions through the center of the journal, leaving the journal at the center of the bearing at a much higher temperature than at the ends. Consequently the journal becomes larger in diameter at the middle of the bearing on account of this expansion and a greatly increased load at that point results.

This variation of heat through the center of the journal is easily discernible in the case of railroad car journals, and an experienced inspector can readily detect the running condition by examining the end of the journal; if any unusual temperatures are being generated, the center of the axle will show a dry condition. When bearings show a tendency to heat, the average man in charge of such work becomes unduly alarmed and as a result goes to extremes to control the heat. About the first thing he thinks of is to apply a heavy oil (generally valve oil) or some heavy grease, slacken off the bearing without even knowing whether or not it is too tight, or apply solid lubricants, though it is apparent that the bearing is not abraded.

Some of the so-called textbooks on lubrication are accountable for this practice, as they state that temperature should not go above 160 deg. fahr. because it is dangerous. They do not hesitate to advise the use of sand, brick dust, etc., and in fact recommend everything or anything except ascertaining the initial cause of the heating and correcting it, which is really the only sane method of curing a hot bearing. In nearly every case of this kind it will be found on investigation that the trouble was caused through some stoppage in the supply of oil to the bearing that can be remedied.

In railroad service there is actually a rather small margin between a bearing that is running normal and one that is running at too high a temperature, and railroad men generally consider that they are getting good results if freight cars make 35,000 car-miles per hot box, passenger cars 175,000 miles, and locomotives 75,000 miles. The locomotive has 32 bearings liable to heat as compared with 8 on the car, but with the extra attention usually given locomotives and passenger cars, better results are naturally expected.

In the author's experience, the greatest number and variety of lubricating troubles have been brought about through carelessness of one sort or another, such as mixing and contaminating the oils, using wrong oils, dirty conditions in the oil rooms and at machines—causing grit and dirt in the oil cans and bearings, failure to fill oil cups, oil being applied at point of bearing load, moisture getting into bearings, steam or gasoline leaking into crankcases, and general lack of attention and repairs to the lubricated parts.

LUBRICATING VELOCITIES OF JOURNALS—BEARING LOADS AND TEMPERATURES

The great majority of bearings (medium-speed journals) on industrial machinery operate at temperatures of from 100 to 130 deg. fahr. with ordinary atmospheric temperatures. Such bearings are considered too hot when their temperature reaches 80 deg. above the temperature of the room where they are situated. This does not apply to the entire range of lubricated bearings as the temperatures of locomotive driving journals and engine-tender and car bearings often far exceed this margin.

At or near the surface of a locomotive bearing the average running temperature is about 240 deg. fahr., on bearings of locomotive tenders and cars, about 180 deg. The author has tested car journals running at 275 deg. and driving journals at 300 deg. without either of them showing any bad results from such temperatures, but does not infer that the temperatures are not too high or that bad results would not take place were they continued, as he simply gives the figures to show that in railroad lubrication the lubricant must have a very considerable factor of safety to meet the unusual conditions constantly encountered. Increased load per square inch due to heavy piston thrusts, slack bushes or bearings, excessively high or variable speeds, unequal load on bearings due to wedges sticking or faulty, damaged equalizing gear, dust and grit from ballast, heavy and high-speed hubbing of driving and engine-truck-wheel hubs, are some of the obstructive conditions peculiar to this service. While no lubricant can be expected to maintain normal running temperatures under these conditions, the very exigencies of the service demand that it be of a quality capable of holding these liabilities under reasonable control, to obviate failures.

The average railroad official is not very much interested in either friction or running temperatures, but what he does demand is a lubricant that will meet successfully occasional or even frequent adverse conditions, eliminate failures on the line, and not cause excess wear of lubricated surfaces in general. Yet consideration of the first-mentioned factors is necessary in the intelligent selection of a lubricant that can deliver and maintain the efficient service thus expected.

The velocities of journals cover an extremely wide range, steam turbines having been built that have a velocity of 30,000 r.p.m., and in textile mills many of the spindles run at 11,000 r.p.m. However, these have a very light bearing load, and, as the velocity at which turbines operate creates a tendency for them to rotate about the gravity axis instead of the mechanical axis, special floating bearings are used in many cases. On railroad equipment the journal velocities range from about 800 to 1200 ft. per min. at a train speed of 60 miles per hour, but in general the loads carried on railroad journals are much greater than those on other classes of machinery.

Freight-car-journal loads run as high as 470 lb. per sq. in., while locomotive driving journals carry a dead load of from 200 to 250 lb. per sq. in., and the maximum thrust load on these is very high. Main crankpins have a thrust load as high as 2000 lb. per sq. in. and many of the journals are often subjected to much greater load for a protracted period due to defects in the equalizing gear, etc. This, together with the variable speeds of trains, dirt and grit picked up, high temperatures due to hub friction, moisture from rain and snow, and changing atmospheric temperatures, combine to make the lubrication of railroad equipment so liable to difficulties that the use of the very best oils procurable for the purpose—irrespective of what the unit cost of the lubricants may be—is not only advisable but absolutely essential to efficient and satisfactory operation.

Bearings may be classed generally under two divisions, journal bearings and thrust bearings. Journal bearings are cylindrical and the load is carried at right angles to the axis of the bearing. When the load is carried or applied in a line parallel to the axis, we have what is called a "thrust" or "step" bearing. In addition, a further class might be added consisting of those bearings which permit moving parts to slide over them. These are known as "sliding-contact" bearings.

But from a lubrication standpoint it is necessary to differentiate further as there are a variety of conditions presented under each of the main classes. A few of these are:

Dead-running journals with uniform load always applied in one direction and a constant journal velocity of over 100 ft. per min.

Bearings where the load is constantly shifting from one side of the bearing to the other, such as the crankpins and main bearings of a reciprocating engine, and where the load is not uniform.

Bearings where the motion is reversible.

Bearings with journal velocities below 100 ft. per min.

Bearings subjected to heat other than that generated by bearing friction.

Bearings subjected to moisture.

Bearings exposed to low atmospheric temperatures.

There are many others, and in almost every case a different class of lubricant is required, and often a different method of application.

METHODS OF APPLYING LUBRICANTS TO BEARINGS

The difficulty presented in supplying proper lubrication to a cylindrical bearing arises from the fact that the lubricant will not flow at right angles to the axis of the journal, nor parallel to the line of surface motion, but has a tendency to flow at various angles from such lines, and always toward the ends.

Such flow action is readily understood when the available pressures on the oil film under the load area at various points on the bearing are taken into consideration. For example, if a row of holes are drilled one inch apart in a bearing parallel to the axis and a few degrees past the center of the load area and are connected to pressure gages, it will be found that the pressure on the oil film is lowest at the outer ends and increases as the center is approached. The center gage will register an amount of pressure per square inch about twice that of the bearing load per square inch. If the row of holes is drilled at right angles to the axis of the bearing (and in the center of bearing length) and connected to pressure gages, the readings will show much the same as the above, with the difference that the minimum pressures are lower and the pressure increases at a more rapid rate toward the center of the load area.

We therefore cannot expect that the lubricant will flow toward the point of highest pressure, and this is demonstrated by a simple illustration. Take a car journal—or any journal—say, 10 in. in length and apply a piece of saturated packing 2 in. wide to the center of this journal when in motion. The result will be that a film of oil will spread all over the journal surface, which shows that the oil has a tendency to flow toward the ends and away from the area of highest pressure. Now remove the waste pad from the center position and place instead a packing 2 in. wide at each end of the journal, and it will be found that the center of the journal cannot be lubricated in this manner.

It must then be evident that on account of the angle followed by the oil flow it is a difficult though highly important matter to lubricant the center of a journal at the point where pressure is the greatest—and this is the particular place where adequate lubrication is most required. Some authorities recommend elaborate designs of grooves as a means of distributing oil to the ends of the bearing. As a matter of fact, there is no necessity of worrying about getting oil to the ends of the bearing, and if any groove is employed it should be one designed to prevent such a flow to the ends and assist in diverting it toward the center of the bearing. It is the author's opinion that in general all such grooving should be angular, with the vertex in the center of the bearing length.

It was formerly the practice on railroads (and in some instances continues to be) to have a groove cut in a bearing parallel to the journal axis and in the center of the load area. The result is that this groove simply breaks the lubricating film at the point where its stability is vital and allows the lubricant to flow to the outer ends of the groove and escape, where the pressures are less. The author was pleased to note the remarks of Professor DeBaufre in a report that he made on a lubrication subject, and while he cannot recall the exact wording, the substance was essentially this:

> No groove was cut in the crown of the bearing parallel with the axis of the journal, as such grooves break up the oil film and allow the lubricant to escape at the ends. However, to provide an auxiliary oil supply on the crown, holes were drilled, staggered over the crown face of the bearing, and these holes held a supply of lubricant at a pressure corresponding with their location.

The author has seen bearings designed which show the oil applied near either end, but none at the center of the bearing. In his opinion this is entirely wrong, for the reasons previously stated. Center application must be used for efficient lubrication. He has many times demonstrated by experience that any bearing, irrespective of its length, can be efficiently lubricated through central application, if proper quantities of the right lubricants are used. The best point for application is at the center of the length of the bearing and at right angles to the line of load, on the outside of the bearing.

Power Losses Due to Frictional Resistance

Probably the principal reason why the lubrication efficiency of oil is not given more general attention is because a great proportion of the losses incurred through faulty lubrication are hidden. As long as an oil is not giving any unusual or flagrant trouble, it is often assumed that it is efficient, while as a matter of fact, it may be causing a heavy power-load factor.

Even where the friction load is known it is often misunderstood, or rather misinterpreted, as to its relationship to other factors. As an illustration of this, at the time of the Federal Administration of Railroads the problem of fuel conservation was given close study and attention. The author was then attached—in a minor capacity—to the New England States Fuel Conservation Committee. He had occasion to take exception to an item on "Friction of locomotives" contained in a pamphlet that the committee had copied from a report issued by the Engineering Department of the University of Illinois. This item was to the effect that the friction of the locomotive consumed 1 per cent of the total coal fired. In a sense the statement is approximately correct, yet when taken at face value, as it was by many railway mechanical men, the loss through friction seems insignificant. But when it is remembered that friction's toll is taken *after* the power is developed at the piston rod and that only about 7 per cent of the total coal fired becomes available as power at this point, the loss through friction is as 1 to 7, or about 14 per cent of the generated horsepower.

Therefore, with the proper ratio of friction to developed power established, it may easily be seen that each reduction of one-fourth in this seemingly unimportant 1 per cent becomes equivalent to an increase of nearly 4 per cent in drawbar efficiency. It is always well to remember that when frictional loads are given in percentage of fuel consumed, the ratio of the friction to the power developed at the piston rod or at the motor must be known, also the percentage of the heat units in the coal developed as power at these points, before the true meaning of friction can be estimated in this way.

Let us also consider the friction of freight-car journals. On a car with a total weight of 80 tons and running at 25 m.p.h. the friction of the journals is approximately 0.995 hp. The total power required to run the car at this speed is about 14.95 hp. with efficiently lubricated journals. The friction of the journals therefore constitutes about 6.6 per cent of the total resistance, but it has been repeatedly demonstrated in tests that there is a vast difference in the coefficient of friction when different oils are used. In these tests a high-grade efficient oil showed a coefficient of friction of 0.0018, while when an inferior oil was substituted the coefficient of friction was 0.0035, or practically double. This difference translated into horsepower shows that it means an increase in horsepower from 14.95 to 15.75, or 0.8 hp. per car. The same ratio of frictional losses is undoubtedly applicable to the bearing surfaces of any machine.

Packing Waste

A great many classes of bearings such as locomotive and tender trucks, passenger and freight cars, mine cars, street cars, electric locomotives, etc., are lubricated by the application of oil-saturated waste. As these comprise a very important division of mechancial equipment, a few remarks on waste lubrication may be appropriate.

Considerable controversy has arisen from time to time as to what kind or class of waste is the most suitable for this purpose. The question is still an open one and those in charge of such equipment continue to use their own judgment in the matter, consequently nearly all kinds of waste are used.

Authorities state that a suitable waste for packing must contain not less than 50 per cent of ply-thread cotton to provide proper capillarity, and that 50 per cent must be a good grade of long-fiber wool to give resiliency and prevent excess glazing on the surface in contact with the bearing.

The author doubts very much whether the capillarity efficiency of a waste is of any great benefit in this class of lubrication, and his reason is this:

The true relative capillarity as between different liquids is tested by tubes, which may be set in either a vertical position or on an angle with their bottom ends in the liquid. Packing waste functions in the same manner, as the spaces betwen the minute fibers form tubes. But if capillary tubes were pliable and were twisted up and rammed into a bearing with perhaps both ends in the liquid or no ends in the liquid, they could hardly be expected to act in the manner it is stated and expected that packing waste will act.

In the care of railroad signal lanterns the company with which the author is connected instructs the users to be very careful in applying the burners to avoid any twists in the wicks, as these twists will prevent the flow of oil in a capillary manner through the wick. Such is actually the case, and it is reasonable to suppose that the principle is as applicable to waste strands as it is to wicks.

A simple test will show that, in the lubrication of car journals with saturated waste, capillarity of waste does assist to some degree, but that it is not the sole agent of such lubrication. Pack a car-journal box with saturated waste and put it into service; after a stated number of miles, even though the packing may have been kept up to the journal by lifting it on the surface with the small hooks used for the purpose, the temperatures will gradually increase, although there is plenty of oil in the box. But this oil will be found at the bottom where it has gradually filtered down through the waste, and will not return to the journal by capillary attraction in sufficient quantities to maintain normal temperatures. Now what is necessary to do again to bring these temperatures back to normal? Simply this: a packing tool must be applied to spade up the packing and force the oil up through the waste, or, in other words, resaturate the waste.

On this account one of the essentials of an efficient lubricant for this class of service is that it shall possess the property of being able to remain in suspension in the waste for the maximum length of time at running temperatures, and the best waste is the kind that will hold the maximum amount of oil in suspension for the maximum time at running temperatures. This does not, of course, refer to any of the patented methods of waste lubrication where the waste is held in a frame in straight strands with one end in the oil and the other in contact with the journal.

The following data regarding the capillarity of different wastes may be of interest.

Kind of waste	Height of lift, in.	Time consumed, min.	sec.
Cotton slasher, tight-spun	2	30	..
Cotton-ply thread	2	1	..
Wool skein yarn	2	6	20
Axminster merino wool	2	0	29
Ingrain merino wool	2	0	30

On a resiliency test the decrease with wool was 25 per cent and with cotton, 46 per cent; these were the resiliency averages when tested with pressures of 140, 70, 30, and 7 grams per sq. in. on an enclosed column of waste 3 in. high.

Preparation of Saturated-Waste Packing

The waste should be pulled apart, tedded out, and put into the oil baths in small quantities; when the required amount has been put in the tank the oil should completely cover the waste. It should then be allowed to soak for not less than 24 hours. The quantities of both oil and waste in the tank should be known, and at the end of the time named the excess oil should be drained off, leaving only 4 lb. of oil for each pound of waste.

In no case should the dry waste be placed in the soaking tank and then oil poured over it, as such a method makes saturation very slow, the air in the waste being all trapped below the oil, making its expulsion slow and difficult. There are many patented machines on the market now by means of which saturation is

accomplished quickly in a vacuum chamber; they are not, however, in general use.

The reason that 24 hours is necessary for proper saturation by the tank-bath method is due to the long time required before the air can be entirely expelled from between the small fibers in the waste and replaced with oil, and the temperature during this process must be about 70 deg. fahr. for complete saturation within the period named. The two most important features of preparation are, first, to know that the waste has been thoroughly saturated; and second, to know that there is a proper amount of oil carried in the waste when it is applied to the bearing. Prepared packing at the time of application should consist of 20 per cent waste and 80 per cent oil by weight, not bulk.

Reclamation of Used Packing and Oil

A considerable number of oil users assume that old oil or oil that has been in use is unfit for further service; they seem to think that it becomes "worn out," or depreciated in lubricating value. It seems scarcely necessary to try to refute this idea, as it has been repeatedly proven to be erroneous. Tests made at the Engineering Department of Cornell University showed conclusively that lubricating oils (that were not built up) did not "wear out" in service, and while the oils gained in viscosity and gravity—due no doubt to the more volatile parts being dissipated by bearing heat—they gave a lower coefficient of friction at low pressures and but a slightly higher one at high bearing pressures.

In railroad and street-car service it is now the general practice to reclaim all old packing, and while there has been some very elaborate and efficient equipment invented for the purpose, the following method is commonly used and is satisfactory:

The old packing is shipped in tight containers to one or more reclaiming plants. It is then pulled apart and foreign matter or short ends that it may contain in considerable quantities are extracted. The packing is then thrown over a screen of about $1\frac{1}{2}$ mesh; this also assists in removing the small foreign substances and short ends of waste. The waste is now placed to a depth of about 3 in. in a vat of oil that is heated to a temperature of about 180 deg. fahr., and shaken up with a fork—or some such tool—for about ten minutes. The depth of the hot oil in the vat is about 16 in. and there is a screen 4 in. from the bottom. The hot oil reduces the viscosity of the old oil in the packing and also expands the waste, freeing it of accumulated grit and dirt; the shaking of the waste causes this foreign matter and any moisture that may be present to fall to the bottom and under the screen. The waste is then removed from the vat and all oil extracted from it by either a centrifugal extractor or a press. This oil is then put through a filter and becomes again ready for use.

From the time new packing is applied until it is re-applied as reclaimed packing, there is a loss of approximately 40 per cent, of which amount about one-half has been consumed in service and the other half discarded as worthless when reclaiming. This loss is made up by the addition of new packing mixed with the old in this proportion. The cost of reclaiming is about 0.62 cent per lb., depending upon the locality. The relative cost of 100 lb. of new prepared packing as compared with the cost of the same amount composed of 40 per cent new and 60 per cent reclaimed is approximately (with cost of new oil 32 cents per gal. and of waste, 14 cents per lb.):

100 lb. New Packing		40 per cent New—60 per cent Old	
Oil, 80 lb.	$3.20	Oil and waste reclamation	$0.45
Waste, 20 lb.	2.80	32 lb. new oil	1.28
		8 lb. waste	1.12
Total	6.00		
	2.85	Total	$2.85
Saving	$3.15		

On railroads and street cars the viscosity of the reclaimed oil should be inspected, especially in the cold seasons, because if the oil has been in service for a considerable time it is liable to increase to a very high viscosity at temperatures below 75 deg. and when found so, should be cut back with a lighter oil such as winter car oil.

The filtration of used oil is a subject that seems to be of more than ordinary interest and one that oil men are constantly being asked about. While there are many methods of filtering, the author believes the following is efficient and will prove satisfactory.

The oil is poured through a screen and then passes through a pipe to a point near the bottom of the precipitating filter where it is heated and the water contained precipitated; the oil then flows to the top of a receiving tank in a zigzag course over horizontal pans which permit the precipitation of the fine particles of foreign matter that it may contain; it is then passed to the cloth filters.

Another type of precipitating filter in general use is one by which the oil travels vertically up and down over partitions in the filter. This method is not to be recommended because it involves the necessity of the dirt contained being precipitated through a high column of oil, whereas in the first-mentioned filter the oil flows in a comparatively thin sheet and the precipitation of dirt is more easily and more thoroughly accomplished.

Some small users of oil filter it through a waste mass instead of the cloth filter. This is a very poor method as there is a great liability of small unseen particles of the waste getting into the oil, and these are very conducive to heated bearings as this material carbonizes on the journal and is often the source of heated bearings on railroad cars.

Other oil users attempt to filter by what is generally known as the "wet" system, which consists of passing the oil through water. The method is practically useless as the oil will pass through the water in drops and inside these drops will still be carried the fine particles of dirt that filtering is supposed to remove.

There is one class of lubricant for which the author cannot recommend any method of filtration, and that is the oil from the crankcases of internal-combustion engines, especially automobiles—unless some qualified engineer or chemist is in charge to ascertain and control the flash point of the oils thus used. He does not know of any method of removing the gasoline from such oil except by redistilling, and the custom of filtering such oil and using it over and over again is dangerous.

Cost of Lubrication

With the possible exception of those applying to railroad equipment, there are few available or reliable data on the actual cost of lubrication per unit of service rendered. In manufacturing plants there is no common basis that would enable one to make calculations that would be acceptable as reasonably accurate approximations. In plants such as textile mills the greater percentage of power is consumed in overcoming friction and maintaining the acceleration of numerous high-speed parts, while in plants on the order of heavy machine works where the greater percentage of power is applied directly for useful work, the friction load of the bearings is low, and ascertaining the cost of lubrication is an entirely different proposition.

Lubrication service is so widely divergent that it would be impossible to establish a standard of efficiency that would be generally applicable. But invariably it will be found that the use of lubricants that can demonstrate the highest service efficiency will, regardless of their necessarily higher price, return the lowest actual unit costs for the work required.

To properly estimate the cost of lubrication, the following data would have to be available:

Quantity of lubricants used
Value of lubricants used
Square feet of lubricated surface rubbed over
The power losses due to friction
The power losses in cylinders and valves due to leakage and poor steam distribution caused by worn parts
The cost of material and labor in making renewals of parts due to unusual friction wear
The delay to machinery while making such renewals.

In a test made not long ago in a manufacturing plant, two grades of oil were used for the lubrication of shafting, machines, etc. With all machinery running light it was found that the power required with the high-grade oil was 203 hp., while with the cheaper-grade oil 210 hp. was necessary. On this basis a run of one week with the higher-priced oil showed that 95 gal. were consumed to maintain power requirements, while with the cheaper oil 112 gal. were needed. The final result of the test showed that the

use of the higher-priced but more efficient oil saved $520 per year in the oil cost alone, without consideration of the savings accomplished through lower depreciation of machinery.

Irrespective of the price of lubricants, or of any specifications that may be used, a really efficient lubricant must meet the following requirements:

1 It must be of sufficient body to keep the bearing surfaces apart at working temperatures
2 It must possess lubrication qualities that will reduce friction to a minimum
3 It must remain fluid at the lowest temperatures that are met with under ordinary service conditions
4 It must meet service requirements as to durability, resistance to elements, usage, etc.
5 It must contain no impurities which would corrode or pit the bearing surfaces
6 It must have no tendency to decompose or form deposits which may gum up the bearings
7 It must be of sufficient capillarity to feed evenly and successfully, with little change in this respect when encountering natural changes in atmospheric temperatures.

These essentials as a guide for selection apply with equal importance to practically every class of machinery requiring lubrication—from the automobile to the modern locomotive. There can be no compromise on quality, nor neglect of the necessary details of care, if really efficient service is to be secured. The author ventures to say that there has never been an instance where skimping on lubrication quality did not prove to be an expensive policy or experiment.

This is particularly true of heavy machinery, where lubricants are called upon to bear extraordinarily weights and strains. For average requirements or even with large consumers, there is really very little money difference between the initial cost of reliable, high-grade oils and those physically unsuited for the work. But there is a much greater difference when comparative service results are analyzed. Any sacrifice in quality to accomplish a tentative saving will invariably result in the user's paying twice over for his lubrication as a penalty for using poor judgment in doing so.

Future of the Machine-Tool Industry

A Forecast Covering the Next Fifteen or Twenty Years, Dealing with Design, Construction, and Business Phases, and Based on Opinions Held by Leaders in the Industry

BY O. B. ILES,[1] INDIANAPOLIS, IND.

THE value of a forecast of any kind depends upon the ability of the forecaster to accumulate reliable data on conditions past and present, to note the effect of certain correlated conditions in the past, to note the present conditions, and by proper deductions to conclude what the combined facts and conditions will produce in the future. The speaker has written to a number of the leaders in the machine-tool industry of the United States asking them their opinion of the future of the industry, not only as to design and construction but as to the business phases in general. The replies have been numerous, and on some points radically different, but on the whole much the same.

For an industry that has contributed so much toward the development of the manufacturing and business interests of the world, the men responsible for the machine-tool industry have not been financially commensurately rewarded. If any large fortunes have been acquired quickly in this industry, the speaker does not know of them. Engineering and designing skill and ability have made possible the development of every factory where metal must be machined. A mechanized civilization is founded on machine tools. Without machine tools it would not have been possible to have developed mass production of the articles that have contributed and are now contributing to our so-called advanced civilization. So it is quite natural that those engaged in the machine-tool industry should inquire what has the future in store for us.

DEVELOPMENT DICTATED BY NECESSITY

The development in machine-tool design in the last twenty-five years has been rapid because of the rapid development of the so-called mechanical age, particularly in the manufacture of metal products during that time. Machine-tool designers have been led somewhat by the requirements and demands of the users during those years. If the users demanded a tool for drilling, milling, turning, boring, or planing, the designer gave him the best that ability, skill, and experience dictated. Tools that a few years ago were considered more or less special are today considered standard machine tools. This transformation is prevalent today and it is reasonable to expect its continuance in the future. Mass production demands larger and more economic production. The advent of high-speed cutting tools demands larger, stronger, and more rigid machine tools. It is reasonable to surmise that the limit of cutting tools has not been reached as yet, and that the future design of machine tools will have to keep pace with the metallurgist's development of cutting alloys.

The more experience a machine-tool manufacturer has with the performance of his product under the many and varying conditions his machines are called upon to function, the more he is able to strengthen the weaknesses, correct the errors, refine the operations, and add to the conveniences of his machine.

IMPROVEMENTS IN DESIGN

It seems reasonable that the machine tool of the near future should come nearer and nearer the perfection we are all seeking and striving for, in spite of the fact that some think we may have reached that stage now. Some of us thought that twenty years ago we had about reached that stage of perfection. Perhaps we were as near perfection then as now, considering the requirements then, now, and what they will be ten or twenty years from now.

The demand and requirements will always to a large degree dictate the design, quality, and productive capacity of machine tools. There has always been a necessary relation between design and development in the metal-working industry and in machine tools. If more strength, more power, greater feeds and speeds, and more convenient operations were required by the user, the machine-tool builder has had to comply with these requirements. Nor have the progressive machine-tool designers and engineers waited for the user to suggest design, but have foreseen most of the requirements and endeavored to supply the users accordingly. It is reasonable to predict that this condition will continue and increase, and that coöperative effort on the part of builders and users of machine tools will grow stronger and closer in the future.

AUTOMATICS

Some of the leaders in the machine-tool industry and many of the largest users of machine tools think that the demand for single-purpose machines, such as semi-automatics and automatics, will become more and more pronounced. In the past, in the United States, the designs and styles of machine tools have been largely determined by the demand. This applies particularly to special-purpose and automatic tools. Inasmuch as the trend in many lines of industry in the immediate past and at the present is toward concentration in large shops and the securing of lower and lower production costs, it is reasonable to forecast an increasing demand for more and better automatic machine tools.

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[1] President, International Machine Tool Co.; President, National Machine Tool Builders' Association.

An address at the New Haven Machine-Tool Exhibition, New Haven, Conn., Sept. 29, 1925, before the New Haven Section of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. Abridged.

If quantities are large enough, this demand will be likely to lead to combination automatic machines, doing multiple operations of milling, drilling, boring, turning, etc. on one machine. This is being tried in a small way now. Opinion of the leading builders is divided on the extent to which a machine of this type can be economically adopted. One leading machine-tool builder says, "Experience has taught both the user and the builder that segregating operations tends to best workmanship and production. Combining operations leads to complications which lower production, due to care of tools and breakdowns." But with more application of inventive and constructive skill, who shall say that the machine-tool designer will never overcome these troubles which now confront him?

STANDARD MACHINE TOOLS STILL NECESSARY

While conditions point to the larger use of automatic and single-purpose machine tools in mass-production shops, there will always be a demand for the so-called standard machine tools—engine lathes, drill presses, shapers, etc.—not only in small-production shops but in the large-production plants. As the automatics are further developed the next decade will see a still further development in standard tools, always meeting the requirements of the future cutting materials in strength and rigidity and advancing over present designs in simplicity of operation, accuracy, and the maintenance of accuracy for the greatest length of time.

EFFECT OF LABOR CONDITIONS

Until recently men anxious and willing to do manual labor have been plentiful, but now their number is decreasing and their labor becoming more and more costly. Our system of education has been making "white-collar" help out of them. Numbers of our large manufacturers have been largely dependent on immigrants for this class of men, but our immigration laws at present prevent them from coming to us, and if such labor cannot be had at a reasonable cost, some substitute must be supplied. What is there in prospect, it may be asked, except machinery, particularly automatic machinery, to supply this deficiency and demand? The machine-tool engineer, designer, and builder should be much in demand during the next decade by the manufacturer of all kinds of machinery and commodities.

NECESSARY REDUCTIONS IN PRODUCTION COSTS

Until now the industries using machine tools have been growing and expanding. New lines have come into existence due to the demand of our people for a higher standard of living. During this period of expansion the methods of pioneers have prevailed, with too little regard for the wastes and refinements in manufacture. In ten or fifteen years from now our present methods in many instances will be considered too wasteful to be tolerated. Today the problems are being studied by engineering brains with a view to eliminating these wastes.

Many thoughtful business men, like A. Lincoln Filene, think we have already entered on a period of most severe competition in manufacture and business, one requiring that production costs in manufacture be reduced to the minimum by mass production by the introduction of every refinement and device that will save time and labor, and by the elimination of all wastes in time and material.

DIFFICULTY OF MASS PRODUCTION

During this period of severe competition the machine-tool designer, engineer, and builder will be in great demand for furnishing high-production machine tools. But this same competition will enter into the machine-tool industry. The machine-tool builders will be compelled to look to the cost of their own products and to their own shop efficiency and management more closely than in the past. The very latest equipment and methods must be employed by them as well as by their customers to reduce cost and eliminate wastes. Owing to the conditions that have prevailed in this industry in the past and are now prevailing, we do not see how it is possible ever to introduce mass production in a machine-tool plant in the same way it is and can be done in plants manufacturing articles for general consumption. The call for special features in machines to meet specific requirements makes nearly every machine tool more or less a special machine and precludes large production. The cost of carrying large stocks with the ever-present danger from obsolescence makes it too hazardous to manufacture to stock even the most standard of machine tools in large quantities. The present tendency of design and manufacture is toward what is called the "standard unit system." This enables the machine-tool builder to manufacture certain parts and units in fair quantities. He designs certain units so that they can be applied to more than one size or style of machine, and can thus be produced in larger quantities and the costs correspondingly reduced. There is but limited demand for large quantities of any one type of machine tool. There are exacting requirements in regard to accuracy and the continued maintenance of accuracy. There are suddenly recurring periods of little demand. There are always new designs, and the bettering of design and development of machine tools to meet ever-changing requirements. As in the past, we think that in the future all these things will preclude mass production of machine tools.

ORGANIZATION

Besides thinking of future designs and styles of machine tools and the shop methods and plans for reducing costs and eliminating wastes, another very large and important part of the industry is to be considered. That is the business end—the form of organization, the future methods and plans for sale and distribution and the ever-present problems connected with the sale of our product at a reasonable profit.

In nearly all industries, in the effort to get the product to the consumer at the lowest reasonable cost—to eliminate all the waste possible—there is evident advantage in combining small establishments into a relatively few large organizations and thus reducing the overhead expense. While the machine-tool industry operates under different conditions from most other industries, here, too, the trend is toward large, more powerful, and more efficient organizations. There have been some combinations of machine-tool builders in the past, and it is reasonable to expect this to continue.

It seems logical to predict more success for combinations that will bring together types of machines representing a single line rather than for organizations that make a diversified line of tools embracing several lines. Years ago one machine-tool shop would make lathes, planers, milling machines, drill presses, and perhaps other types of machines. Gradually most of these shops centered their efforts on one or two of these lines, with better results not only for themselves but for the trade. They gave all their thought, skill, and ability to higher-grade production in one line rather than distributing it over several lines, and specializing became more and more in order. But with the increased demand for more production it seems logical to expect specialization to develop from now on along the line of a given art in the machining of metal rather than on a given type or size of machine within that art. Such organization will give the advantage of larger-scale production, lower selling costs, and better engineering and organization than the more narrow specialization of the past.

A PLAN FOR THE SMALL MACHINE-TOOL SHOP

But there will always be the small machine-tool shop. There will always be mechanics and engineers who do not work best in subordinate positions, who have worth-while ideas and designs and sufficient persuasive power to get financial backing to build and market machines embodying them. One of the great faults of large organizations is the submerging, and sometimes the loss, of individual responsibility and effort. Unless this individual effort is cultivated and maintained in any large organization, the utmost success will not be realized and failure is probable. The advantage gained by size, ability, and means is more than offset by the lack of individual coöperative effort and interest. The future large machine-tool organization must find a way to get and maintain this individual coöperative interest and effort among its staff.

SALES POLICY

One of the most important items for consideration in the machine-tool industry is the sales policy. The method of selling machine tools

by most manufacturers in the United States and other countries has been through machine-tool agencies or dealers. This is particularly true of the smaller and medium-sized manufacturers. Some of the larger concerns have sold their product directly to the users for years. The present tendency is more and more toward direct sales. This is particularly true of the large organizations and builders of more or less special machines. If we are right in our opinion that the machine tools of the future will be made by larger combined and concentrated organizations, it is reasonable to believe that they will tend to direct selling.

For a time the machine-tool dealer handled only a limited number of non-competing lines. He and his salesmen were able to familiarize themselves with the design, purpose, and productive capacity of the tools they were selling. Today many dealers in machine tools have entirely too many lines to do justice to any of them. Under such circumstances salesmen often do not even recognize the class of work a certain type of machine tool is intended to perform, with the result that the manufacturer must be content to be inadequately represented or furnish the salesman to sell the machine.

To be successful and efficient, the machine-tool salesman of today must be trained in the design, purpose, efficiency, quality, performance, and productive capacity of the machine tool he is offering for sale. He must be able to advise intelligently with the mechanical and financial departments of the prospective customer in an educational way. He should have the ability to prove the advantage, in dollars and cents, of an investment in his particular tool, and should be able to show the prospect the loss resulting from the use of obsolete machines and methods.

Salesmen of this type can best be trained by an organization that has specialized in a particular art in machine building and machining, turning, boring, planing, or milling metals. As these organizations gather weight by experience along their special line and expand either individually or in combination with others, they should naturally develop the ability and the training methods for producing machine-tool salesmen, engineers, and educators of high quality.

World Markets for Machine Tools

So far we have had reference to the market for machine tools in the United States—the home market. Along with the care and further development of our home market we must give even more attention than we have in the past to the world markets for machine tools. We must apply the same system of education on machine tools to the world markets. We must show them that, regardless of price, an investment in machine tools made in the United States will pay the largest dividends in quality and quantity production.

We may take for granted that there lies just ahead of us a period of severe competition, and that the American machine-tool builder will be a large factor in enabling the manufacturers of the commodities of life to meet this competition by furnishing machines to increase production, save man power, and save waste. Then these same tools should be in demand wherever there is industrial expansion due to civilization's cry for more and more of the commodities of life.

In most of the industrialized European countries, due to the cheap labor available, manufacturers have not been forced to use machinery instead of man power to the extent we have in the United States. Nor have they undertaken mass production to the extent we have. But this will come, and come soon, and along with it will come their demand for rapid-production machines. Japan is already advancing in industry and manufacturing. We hardly imagine the next decade will see great progress in Japan toward the development of industry and manufacturing as we think of it, but it will come. The same is true in regard to a reorganized Russia and Mexico.

Our immigration laws now forbid the coming to us of an unlimited number of immigrants from the industrial centers of Europe. These same peoples have heretofore helped to settle and build up this country industrially. Will they now go to the south-temperate zones and repeat in the course of time what has happened industrially in the United States? Will these people go to South America, South Africa, and Australia? Already Brazil and Argentina are claiming advancement along industrial lines. It remains to be seen whether they will discover the fuel and raw materials necessary to very great industrial advancement. But be that as it may, a higher plane of living is spreading over the whole world, reaching slowly but surely into every quarter and demanding more of civilization's so-called necessities and luxuries of life. All these developments will require machine tools in proportion to the advancement of the plane of living and the accumulation and distribution of wealth by the peoples in these various countries.

Problems Confronting the Association

One of the problems before the National Machine Tool Builders' Association is the reduction of the expense, loss, and waste caused by periodic slack demand for machine tools. It is either a feast or a famine in the demand for machine tools. The users wait until they must have tools before they place their orders. These slack periods compel the builder to break up his organization and take losses that must be made up in busy times, if he is to stay in business.

This is a waste that we wish to overcome. We wish in some way to educate the user to buy his machinery more regularly, to forecast his needs better and not swamp the machine-tool builder with boom-time orders. This is in the interest of economy, efficiency, workmanship, and price. The next decade should see marked improvement in education and results along this line.

Another one of these problems is the disposition of used tools. This problem is growing. A plan needs be worked out whereby economic scrap will promptly go to the melting furnace and yet whereby the tools that have residual value can be intelligently allotted to places where their service will be greatest. While the machine-tool builder is demonstrating that money and time and effort are being wasted and lost by the use of obsolete machines, he must also be able to advise soundly as to the disposition of displaced obsolete machines. It does not necessarily follow that a machine that has become obsolete on one operation or job is obsolete on all jobs. There is often a place for such a machine where its service will be effective during its life and accuracy. The future should bring a more satisfactory plan for the disposition of old, used, and obsolete machine tools than we have at the present time.

Research

Most machine-tool builders are continually experimenting and doing a certain amount of research work on their own initiative and responsibility. There has been little exchange of the results obtained from such research work. There is a vast amount of this research work of a general nature that can be better and more cheaply done by coöperative effort under the direction of an organization like the National Machine Tool Builders' Association and in coöperation with other associations, societies, and schools seeking like information.

It is the hope of the National Machine Tool Builders' Association that in the near future this association will be able to coöperate in research with such laboratories and that the information thus obtained will enable us to hold our reputation as designers and builders of the best machine tools. We need research that will lead to a higher development, better refinement, and larger productive efficiency of all machine tools. But this requires coöperative effort. While such work would be educating the user and ourselves technically and mechanically, it would also educate more of the machine-tool builders of the United States as to the benefits and dividends to be had from a whole-hearted coöperation in behalf of the industry as a whole. In coöperative research any industry learns that it pays well not merely to contribute financially, but to contribute time and effort toward raising the standard of mechanical efficiency of its product, to raise the standard of morals in the industry's business dealings, and so to help in making the world a better place for human beings to live in.

History is being made in this epochal age in the machine-tool industry as well as other industries. Whether the historians record an orderly and sound progress in our industry depends upon its leadership now and in the near future. It is our hope and belief that this leadership will assume the responsibility and justify its position in this world advancement.

Correction to Boyle's Law for High-Pressure Gas

Description of Apparatus and Method for Determining the Deviation of Natural Gas from Boyle's Law, and of Methods for Applying Correction Factors to Meter Readings

By D. E. SILCOX,[1] WHITTIER, CAL.

BOYLE'S law, although stating only an approximation of the volume-pressure relation of a gas, finds its greatest application in the natural-gas industry. Hundreds of millions of cubic feet of gas are metered daily under a wide range of temperatures and pressures, and the calculations involved in reducing the gas volumes to standard conditions of temperature and pressure (usually 60 deg. fahr. and 4, 8, or 10 oz. above a stated barometric pressure) are all based on the assumption that Charles' and Boyle's laws are true.

That natural gas and other gases do not obey Boyle's law has been known for years. Boyle himself was the first to point out that there would be deviations from the law. Several men have investigated the compressibility of gases, but of special interest to the gas industry is the work of G. A. Burrell and I. W. Robertson of the Bureau of Mines, who determined the compressibility of several samples of natural gas and of the pure constituents of natural gas in 1916 and 1917. At about the same time R. F. Earhart and S. S. Wyer,[2] at Ohio State University, made determinations of the deviation of natural gas from Boyle's law. The work of these men showed that at 400 lb. pressure a natural gas with a high ethane content might deviate from the law by more than 18 per cent.

In spite of this evidence of the invalidity of Boyle's law and its importance to the industry, no application of a correction for this deviation was made on gas measurements until a few years ago.

It was noted that in the handling of natural gas there would be considerable discrepancies where gas was measured at a low pressure and again measured at high pressure after compression, and that the plant losses appeared to be excessive. It was known that this was due to the fact that Boyle's law was only an approximation of the actual behavior of natural gas. An endeavor was made by chemical analysis to apply a correction to Boyle's law as used in gas measurement. This after considerable work was found to be impractical. Ross McCollum then devised a system of volume measurements whereby the deviation from Boyle's law could be determined and applied. This deviation also when applied to long transmission lines will show the true line losses when the gas at the different points is measured under different pressures.

Mr. McCollum's method consisted of filling a bottle or cylinder with gas, noting the pressure, and then metering the gas through a test meter. The experiment was then repeated, using air at the same pressure. The ratio of the gas to air volume gave the correction factor.

Of course this method did not give the true deviation from Boyle's law, because the air deviates also therefrom. The reason for employing air was that the orifice plates used for metering gas had all been calibrated with air.

[1] Producing Department, Standard Oil Company of California.
[2] Trans. A.S.M.E., vol. 38 (1916), p. 285.

FIG. 1 APPARATUS AS USED IN TESTS
(Showing gas cylinder in water bath with pressure gage and test meter connected; also water prover and deadweight tester.)

Determinations of the deviations of many high-pressure gases were made several years ago, using the method just outlined, and these corrections applied to gas measurements. This was the first instance where a correction for deviation from Boyle's law was applied to commercial gas measurements.

Later L. G. Gates, who had assisted Mr. McCollum, pointed out that the orifice-plate calibrations had been made at atmospheric pressure and for that reason, in making the deviation determinations the theoretical volume of gas that the cylinder would hold at the pressure of the experiment as calculated by Boyle's law should be used instead of the metered volume of air for comparison with the metered volume of gas. (In all calculations the residual gas in the cylinder is taken into account.)

Since that time the true deviations from Boyle's law have been determined and used in gas calculations. Refinements in apparatus and methods have been made as the work progressed, but essentially the method now used is the same as at the beginning.

The most important improvement made on the method used by Mr. McCollum is that the pressure gage is so attached to the apparatus that a series of readings can be taken and the entire deviation curve determined in one experiment, instead of determining a single point as formerly done. This not only greatly reduces the time of making the determination and reduces the calculations involved to a minimum, but greatly increases the accuracy of the method.

The apparatus used for making the determinations consists of:

1 A carbon dioxide cylinder of 1.2075 cu. ft. capacity (plus fittings and gage, 1.2094 cu. ft. total capacity)

2 (*a*) A wet test meter capable of indicating volumes of 0.001 cu. ft.
(*b*) A meter prover of 0.1000 cu. ft. capacity for checking the test meter

3 (*a*) Three pressure gages. These gages have 12½-in. dials. One has a range of 0 to 25 lb. and the scale is graduated in eighths of a pound. The range of the second is 0 to 400 lb., and scale graduations are at 2-lb. intervals
(*b*) A mercury column for calibrating the low-pressure gage
(*c*) A deadweight tester for calibrating the high-pressure gages

4 A mercury barometer. It is essential that the barometer be correct, otherwise the determined deviations will be low

5 A water bath for the gas cylinder. The water is agitated by permitting compressed air or gas to bubble through it, thus maintaining a uniform temperature throughout the bath

6 Several mercury thermometers.

Fig. 1 shows the gas cylinder in the water bath with pressure gage and test meter connected; also meter prover and deadweight tester.

The capacities of the cylinder and fittings were determined by noting the volume of water required to fill them. The capacities of the pressure-gage springs were determined by filling them with high-pressure gas, noting the pressure and then metering the gas at atmospheric pressure. From the volume of gas metered, the barometer reading, and the pressure the capacity of the spring gage was calculated by Boyle's law. Any error here due to deviation from Boyle's law is negligible, since the capacity of the spring is approximately only 0.001 cu. ft.

The wet test meter is tested at the beginning of each determination with the meter prover, and sufficient water added to or removed from it so that it is reading correctly. In checking the meter the air is run through it at the same rate as the gas in an experiment.

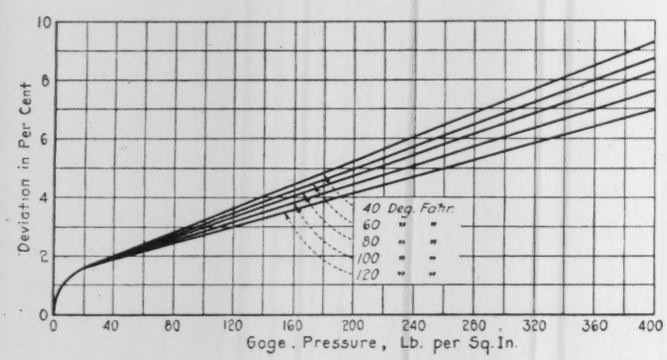

FIG. 2 DEVIATION OF NATURAL GAS FROM BOYLE'S LAW WITH PRESSURE AND TEMPERATURE
(For analysis of gas see Table 1.)

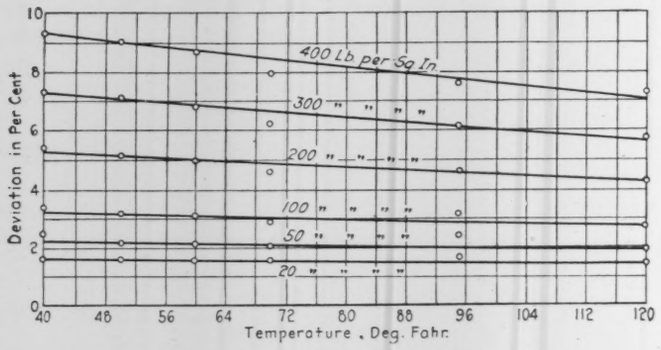

FIG. 3 DEVIATION OF NATURAL GAS FROM BOYLE'S LAW WITH PRESSURE AND TEMPERATURE
(Same gas as in Fig. 2. See analysis in Table 1.)

The gas container is purged several times by filling and blowing to air some of the gas to be tested before the sample for the experiment is taken.

The cylinder is then placed in the water bath and the water brought to the temperature at which it is desired to run the experiment. (It is desirable that the room temperature be near that of the water bath.) The pressure gage is connected to the bottle and the gas turned into it. The meter is set at zero and connected to the discharge valve on the cylinder with a piece of rubber hose.

When the temperature of the gas in the cylinder is the same as that of the water bath, as is indicated by a constant pressure reading of the pressure gage, the initial pressure and meter readings are recorded in columns 1 and 2 of the data sheet. Then the needle valve connecting the cylinder and meter is cracked open, permitting the gas to flow through the meter at a rate of 0.20 cu. ft. per min. As the pressure in the cylinder decreases the rate of flow through the meter decreases. When the rate becomes 0.16 cu. ft. per min., the valve is opened a little more until the rate is again 0.2 cu. ft. In this way the average rate is maintained at 0.18 cu. ft. per min.

The data taken during the test consist of simultaneous readings of both the pressure gage and the wet test meter taken for every cubic foot of gas passing through the meter. Readings of the temperature of the gas passing through the meter are taken every 0.2 cu. ft., and barometer readings are taken every cubic foot. The gas temperatures and barometer readings are recorded on a separate sheet and averaged for computing the correct volume of gas passing through the meter.

The gas is run out of the cylinder until the pressure has reached about 25 lb., at which point the experiment is stopped and the 25-lb. gage substituted for the high-pressure gage. This procedure admits of fairly accurate pressure readings at low pressures. The gas lost in changing gages may be neglected or may be calculated and the meter set ahead that much. From this point on more frequent readings of pressure gage and meter are taken.

At the end of the run the high-pressure gage is calibrated with the deadweight tester, and the low-pressure gage with the mercury column.

The temperature of the water bath should be maintained within plus or minus two-tenths of a degree of the temperature of the experiment.

The calculations are as follows:

Column 1 of the deviation-factor data sheet contains the recorded pressure readings and column 2 the corresponding meter readings.

Column 3 gives figures which are the average temperature of all the gas passing through the meter up to the corresponding meter reading in column 2.

Column 4 contains the average barometric pressure at which all of the gas was metered up to the corresponding meter reading.

Column 5 gives the true gage pressures, which are obtained from the pressure-gage calibration data obtained at the end of the test.

Column 6 gives the true absolute pressures, which are obtained by adding the value of barometric pressure, taken at the time the corresponding reading in column 1 was taken, to the value given in column 5.

Column 7 gives the meter readings corrected to a barometric pressure of 29.92 in. of mercury and the temperature of the experiment. The calculation is as follows:

$$[7] = \frac{[2] \times (T_b + 460)}{[3] + 460} \times \frac{[4]}{29.92}$$

where the numbers in brackets are the values given in the columns of the data sheet similarly numbered and T_b is the temperature of the water bath, deg. fahr.

Column 8 is the corrected volume of gas in the cylinder at any set of readings. It is obtained by adding to the total volume of gas (corrected) through the meter the corrected volume of the residual gas in the cylinder, and then deducting the corresponding reading in column 7.

Column 9 is unity plus the deviation from Boyle's law of the gas being tested. It is obtained by dividing the actual volume of gas in the container (i.e., the figure in column 8) by the theoretical volume of gas that the cylinder should hold. The theoretical volume equals

$$\frac{(G + B)V}{14.69}$$

where G = true gage pressure
B = barometric pressure
$G + B$ = column 6 figure, and
V = volume of the gas cylinder, fittings, and gage
= 1.2094 cu. ft.

The entire calculation for column 9 simplifies to:

$$[9] = \frac{[8] \times 12.1465}{[6]}$$

where $12.1465 = \dfrac{\text{Standard barometric pressure}}{\text{Capacity of the gas cylinder}} = \dfrac{14.69}{1.2094}$

The figures in column 9 are then plotted against gage pressures. The resulting curve gives the correction factors to be applied for any pressure, within the range of the experiment, where the gas is being metered through displacement or volumetric meters. The gas volume is calculated in the usual way and then multiplied by the correction factor.

For orifice meters or flow meters it is necessary to take the square root of the correction factor to be used. A second curve may be plotted by taking several factors off the original curve and ex-

TEMPERATURE AND BAROMETER SHEET

Data for Fig. 4 Date.....192..

Meter	Temp.	Barom-eter	Meter	Temp.	Barom-eter	Meter	Temp.	Barom-eter
1	73.0		46	73.1		91	72.9	
2	73.0		7	73.0		2	72.9	
3	73.0		8	73.0		3	72.9	
4	73.0		9	73.0		4	72.9	
5	73.0	29.025	50	72.9	29.040	5	72.9	29.035
6	72.8		1	72.9		6	72.8	
7	72.8		2	72.9		7	72.8	
8	72.8		3	72.9		8	72.8	
9	72.8		4	72.8		9	72.8	
10	72.8	29.025	5	72.8	29.040	100	72.9	29.035
1	72.8		6	72.7		1	72.9	
2	72.8		7	72.6		2	73.0	
3	72.8		8	72.7		3	73.0	
4	72.8		9	72.7		4	73.0	
5	72.8	29.025	60	72.7	29.035	5	73.1	29.035
6	72.9		1	72.7		6	73.2	
7	72.9		2	72.7		7	73.2	
8	72.8		3	72.7		8	73.2	
9	72.7		4	72.7		9	73.2	
20	72.5	29.025	5	72.7	29.035	110	73.2	29.035
1	72.4		6	72.7		1	73.2	
2	72.3		7	72.7		2	73.2	
3	72.3		8	72.5		3	73.2	
4	72.3		9	72.5		4	73.2	
5	72.3	29.025	70	72.6	29.035	5	73.2	29.030
6	72.3		1	72.5		6	73.4	
7	72.3		2	72.6		7	73.4	
8	72.3		3	72.6		8	73.4	
9	72.4		4	72.6		9	73.5	
30	72.5	29.030	5	72.6	29.035	120	73.5	29.025
1	72.5		6	72.6		1	73.5	
2	72.7		7	72.5		2	73.8	
3	72.8		8	72.5		3	74.0	
4	72.9		9	72.5		4	74.0	
5	73.0	29.030	80	72.6	29.035	5	74.0	29.025
6	73.2		1	72.7		6	74.1	
7	73.2		2	72.7		7	74.1	
8	73.3		3	72.8		8	74.1	
9	73.4		4	72.8		9	74.1	
40	73.5	29.035	5	72.8	29.035	130	74.1	29.025
1	73.5		6	72.9		1	74.0	
2	73.5		7	72.9		2		
3	73.5		8	72.9		3		
4	73.3		9	72.9		4		
5	73.1	29.040	90	72.9	29.035	5		

DEVIATION FACTOR DATA

Gas data for Fig. 4 Date.....192

Time	9.05 A.M.	11.40 A.M.
Room temperature, deg. fahr.	78	79
Water-bath temp., deg. fahr.	80	80
Barometer, in. Hg	29.025	29.025

Volume of container, 1.2094 cu. ft.

Readings: Gage pressure, lb. per sq. in.	Meter reading, cu. ft.	Calculations: Avg. temp. of gas, deg. fahr.	Avg. barom., in Hg	True gage pressure, lb. per sq. in.	True absolute pressure, lb. per sq. in.	Corrected meter reading at 29.92 in. Hg, and 80 deg. fahr.	Vol. of gas in container at 29.92 in. Hg and 80 deg. fahr.	Deviation plus one (1)
290.8	0.000			287.0	301.25		26.966	1.0873
279.8	1.000	73.0	29.025	275.9	290.15	0.983	25.983	1.0877
269.3	2.0	72.9	29.025	265.6	279.85	1.966	25.000	1.0851
259.3	3.0	72.9	29.025	255.6	269.85	2.949	24.017	1.0811
249.0	4.0	72.8	29.025	245.5	259.75	3.939	23.033	1.0771
238.0	5.0	72.7	29.025	235.0	249.25	4.917	22.049	1.0745
227.4	6.0	72.7	29.025	224.3	238.55	5.900	21.066	1.0726
216.8	7.0	72.7	29.025	213.5	227.75	6.884	20.082	1.0710
206.0	8.0	72.8	29.025	202.7	216.96	7.866	19.100	1.0693
85.4	19.0	72.8	29.035	82.6	96.86	18.687	8.279	1.0382
73.9	20.0	72.8	29.035	71.3	85.56	19.671	7.295	1.0356
62.0	21.0	72.8	29.035	60.0	74.26	20.654	6.312	1.0324
50.3	22.0	72.8	29.035	48.5	62.76	21.638	5.328	1.0312
37.8	23.0	72.8	29.035	37.3	51.55	22.621	4.345	1.0238
24.1	24.0	72.9	29.030	25.7	39.95	23.596	3.370	1.0246
	Changed Pressure Gages							
22.85	24.268	72.9	29.030	22.83	37.08	23.860	3.106	1.0175
21.46	24.375	72.9	29.030	21.42	35.67	23.965	3.001	1.0219
18.04	24.674	72.9	29.030	18.04	32.29	24.259	2.707	1.0183
16.62	24.792	72.9	29.030	16.69	30.94	24.377	2.589	1.0164
14.96	24.945	72.9	29.030	14.73	28.98	24.525	2.441	1.0231
13.08	25.104	72.9	29.030	12.98	27.23	24.682	2.284	1.0188
10.50	25.324	72.9	29.030	10.50	24.75	24.898	2.068	1.0149
9.05	25.460	72.9	29.030	8.84	23.09	25.032	1.934	1.0174
7.51	25.588	72.9	29.030	7.43	21.68	25.158	1.808	1.0130
5.00	25.800	72.9	29.030	4.97	19.22	25.366	1.600	1.0112
3.80	25.900	72.9	29.030	3.84	18.09	25.464	1.502	1.0085
2.69	25.985	72.9	29.030	2.82	17.07	25.548	1.418	1.0090
1.32	26.099	72.9	29.030	1.56	15.81	25.660	1.306	1.0034
0.41	26.178	72.9	29.030	.64	14.89	25.738	1.228	1.0017
Zero	26.234	73.0	29.030	0.00	14.251	25.793	1.1732	1.000

tracting their square roots and plotting these values against pressures. Then to correct an orifice-meter reading for deviation from Boyle's law, make the calculation in the usual way and multiply by the proper factor obtained from the second curve.

In making the deviation determinations, at least two tests are run on each gas as checks. If checks are not obtained with the first two tests, additional runs are made until check tests are procured.

The time required to complete a single test is from eight to ten hours. This includes procuring the sample, checking the meter and gages, running the test, and making all calculations. One man can perform the entire experiment, but better results can be obtained where two men do the work. When two men are working together, columns Nos. 3, 4, and 8 can be calculated during the test, and besides better readings obtained. A calculating machine is used for all calculations and reduces the work involved in making the computations to a minimum.

The temperature at which the experiment is run is determined by the temperature at which the gas is being metered in the field. Originally, all tests were conducted at the same temperature, it being assumed that a temperature difference of 10 or 20 deg. fahr. would have little effect on the deviation.

Later tests were run with the apparatus described in this paper

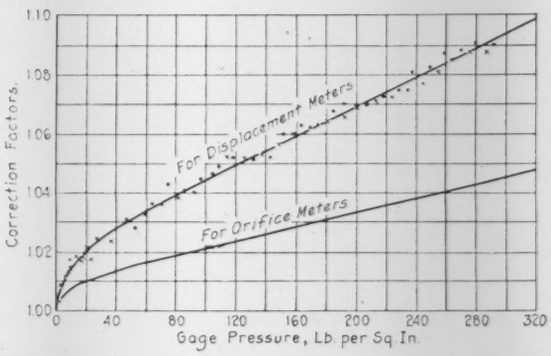

FIG. 4 EFFECT OF GASOLINE CONTENT OF A GAS ON ITS DEVIATION FROM BOYLE'S LAW
(For analysis of gas see Table 1.)

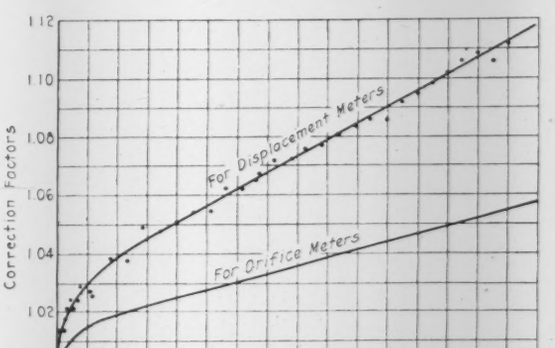

FIG. 5 EFFECT OF GASOLINE CONTENT OF A GAS ON ITS DEVIATION FROM BOYLE'S LAW
(For analysis of gas see Table 1.)

at different temperatures. Figs. 2 and 3 show how the deviation varies with the temperature. It will be noted that through the temperature range 40–120 deg. the deviation variation with temperature is a straight-line function.

Figs. 4 and 5 illustrate the effect of the gasoline content of a gas on its deviation. Fig. 4 is for a gas containing less than one-tenth of a gallon of gasoline per 1000 cu. ft., while Fig. 4 is for the same gas plus the vapor of approximately one gallon of 95-deg. A.P.I. gasoline per 1000 cu. ft. In other words, Fig. 5 is for the wet gas supplied to an absorber plant and Fig. 4 for the dry gas from the same plant.

It is interesting to note in this connection how the addition of approximately 3 per cent of gasoline vapor to the gas of Fig. 4 changes its analysis. It is for this reason—that is, that the methane and ethane content of a gas as determined by ordinary analysis are not methane and ethane only, but also propane and butane and higher hydrocarbons—that analytical methods for determining the deviation of a gas are impractical.

This is further illustrated in Figs. 6 and 7 and their analyses. When the sample for Fig. 7 was taken there was an air leak on the vacuum line to the plant, and as a result the discharge gas was diluted with 12.8 per cent of air. The ethane content of this gas was

24.8 per cent. After the air leak was repaired and the air content of the discharge gas reduced to less than one-half of one per cent, the ethane and methane content did not increase in the same proportion as expected, but there was an actual decrease in the ethane content with a consequent larger increase in methane. This is due to the fact that a small percentage of the higher hydrocarbons has a very great effect on the gas analysis, and that actually the absorber plant was operating more efficiently when the gas was not diluted with air, the gas being considerably drier as shown by charcoal tests.

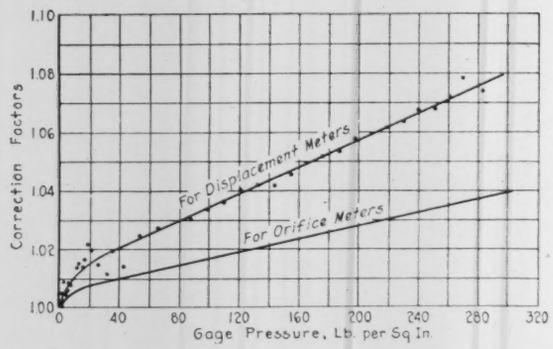

FIG. 6 EFFECT OF GASOLINE CONTENT OF A GAS ON ITS DEVIATION FROM BOYLE'S LAW
(For analysis of gas see Table 1.)

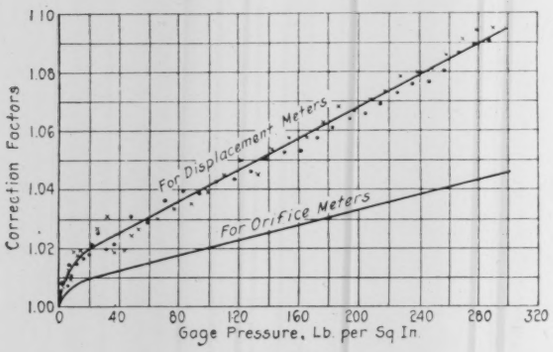

FIG. 7 EFFECT OF GASOLINE CONTENT OF A GAS ON ITS DEVIATION FROM BOYLE'S LAW
(For analysis of gas see Table 1.)

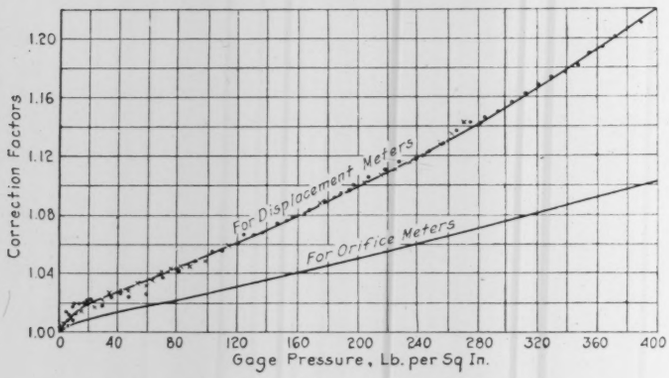

FIG. 8 DEVIATION OF CARBON DIOXIDE GAS FROM BOYLE'S LAW

The heating values given in Table 1 for the different gases were determined with a Junker calorimeter and are gross values per cubic foot of gas at 60 deg. fahr., and an absolute pressure of 14.65 lb. per sq. in.

Specific gravities were determined with an Edwards gas balance.

It will be noted that the correction factors at even relatively low pressures amount to one or two per cent. This is of importance to distributing companies who sell gas at a pressure of 50 to 100 lb.

The importance of these deviation curves is also apparent when one considers compressor-plant operations. If a plant takes gas in with a slight intake pressure and discharges at a high pressure, by not applying any deviation correction it is easy to have approximately 5 per cent plant loss. If the deviation correction is applied, however, the loss will be about 1.0 per cent, or approximately 20 per cent of the former value. This smaller loss is a much more reasonable figure.

TABLE 1 DATA ON THE GASES TESTED

Gas	CH_4	C_2H_6	CO_2	O_2	N_2	Sp. G.	Heating value, B.t.u. per cu. ft.	Remarks
Fig. 2	91.9	1.1	7.0	0.0	0.0	0.626	...	Same gas as Fig. 3
Fig. 3	91.9	1.1	7.0	0.0	0.0	0.626	...	Same gas as Fig. 2
Fig. 4	57.5	21.6	16.0	0.5	4.4	0.834	956	Fig. 5 gas as discharged from absorber
Fig. 5	44.6	36.0	15.4	0.5	3.5	0.895	1080	Fig. 4 gas plus gasoline vapor
	64.7	31.0	1.0	1.0	2.2	0.727	1194	
	66.4	22.9	2.0	1.2	7.5	0.720	1068	
Fig. 6	59.7	24.8	0.1	2.7	12.7	0.743	1034	Fig. 7 gas diluted with air
Fig. 7	73.8	24.1	0.3	0.1	1.7	0.681	1163	
	82.8	12.0	2.3	0.3	2.6	0.646	1041	
Fig. 8	....	....100.0	..	..	.....	....		Commercial carbon dioxide

Line-loss figures are incorrect unless a correction for the deviation from Boyle's law is applied to meter readings. In cases where no correction is applied it usually means that the gas-producing company is bearing part of the burden of the distributing company's line loss and that the distributing company does not truly know what its real line loss is.

Making Pure Iron Commercially

IRON of such purity that it is difficult to analyze for the metalloids, is being produced at the plant of the Niagara Electrolytic Iron Co., Niagara Falls, N. Y. The material is made by depositing the iron upon a steel mandrel through the use of ferrous chloride as an electrolyte with anodes of blast-furnace iron. The tubes so produced are separated from the mandrel by a kneading process. The tubes may then be converted into strips or other forms.

Deep stamping to a degree not possible with soft steel without excessive annealing costs can be performed with this material. The material is therefore suitable for such work as hub caps for automobiles, oil cans, cartridges, vacuum bottles, bottles for ammonia and other gases, metallic hose, etc. It is also claimed that electrolytic iron may be used for making superheater tubes in competition with charcoal-iron tubes and likewise for evaporator tubes to carry alkalies and various salts. The product is said to be dense and of high heat conductivity, great malleability, and resistance to corrosion.

The process was essentially developed in France. French design is responsible almost wholly for the stripping machine, which the mandrel and its deposited iron reach by means of an appropriate electric truck immediately after the primary annealing. Essentially, the machine is a means for rotating the mandrel at definite speed, while at the same time subjecting it to the cold-rolling action of three rollers, 120 deg. apart, each pressed radially against the mandrel by its own hydraulic cylinder.

In the process of stripping it has been found imperative to work upon the central portion of the tube first, leaving about 6 or 8 in. at each end untouched by the rollers. This is based upon the fact that the roller action expands the metal, for if the ends were loosened first, the expansion would take place longitudinally and it would be difficult to strip the tube from the mandrel.

After the central portion is thus stripped, the ends are handled in the same manner, when an elongation of the tube of $\frac{1}{2}$ in. to $1\frac{1}{2}$ in., with an average of perhaps $\frac{3}{4}$ in. to 1 in., is found to have occurred. The action is that of moving rolls outside the tube, with a fixed roll inside, i.e., fixed with regard to the tube itself.

As delivered from the stripper the tube is hard and brittle—so hard, in fact, that it is difficult to cut off a section as a sample for analysis. This condition is overcome by passing it through an electric furnace, giving it the final annealing treatment. The tube passes through the furnace in continuous slow motion, which requires approximately 15 min. from entrance to exit. The tube moves on rollers operated by rope drive from a motor, with appropriate gearing, having reached these rollers through an automatic table which delivers each tube to the rollers just a few inches from the rear end of the one next preceding. Delivered sidewise on to the roller tables, the tube goes lengthwise through the furnace. (*The Iron Age*, vol. 116, no. 11, Sept. 10, 1925, pp. 675–679)

A System of Limits for Different Kinds of Fits

Hole vs. Shaft as Basis of Fit System—Best Method of Expressing Tolerances—Newall and A.E.S.C. Systems—Standard Method Needed to Express Tolerances

By C. D. ALBERT,[1] ITHACA, N. Y.

IN ENGINEERING production three kinds of accuracy are recognized: accuracy of size, of form, and of position. All three kinds are necessary for interchangeability in an assembled product. That mating parts may function properly, certain kinds of fits are found necessary. The Newall system of fits, still used to a considerable extent in England and her colonies, recognizes six kinds of fits. The first report of the Sectional Committee on the Standardization of Plain Limit Gages for General Engineering Work, appointed under the procedure of the American Engineering Standards Committee and sponsored by The American Society of Mechanical Engineers, recognizes eight kinds of fits. The recognized kinds of machined fits under these systems are given in Table 1.

TABLE 1 NEWALL AND A.E.S.C. SYSTEMS OF MACHINED FITS

NEWELL	A.E.S.C.
Easy Running Fit	Loose Fit
Medium Running Fit	Free Fit
Close Running Fit	Medium Fit
Push Fit	Snug Fit
	Wringing Fit
Drive Fit	Tight Fit
Force or Shrink Fit	Medium Force or Shrink Fit
	Heavy Force or Shrink Fit

These various fits require accuracy of size and of form. For each kind of fit the mating parts must differ in size by a certain amount. The necessary difference in size to obtain the desired fit is known as *allowance*. To have an exact allowance for any given fit would require that each part be absolutely true to a predetermined size. As this is commercially impossible, the best that can be done is to stipulate a minimum and a maximum size for each mating part. The minimum and maximum sizes designated for a part are known as the *limiting sizes*. The difference between the two limiting sizes is known as the *tolerance*, and is a means of specifying the degree of accuracy of size required. The practical attainment of an acceptable allowance for a given fit will therefore depend on the tolerances or limiting sizes set for each mating part.

TOLERANCES

Before the limits can be set for a system of fits covering a wide range of sizes, it is necessary to fix upon practical tolerances. The smaller the tolerance the better must be the finish of the surfaces to be measured, and the higher will be the cost of making the part to size within the limits set. Hence, for economy of production, tolerances for the mating parts should be as great as the nature of the fit desired will permit. It therefore follows that the permissible variation of the allowance depends upon the fit desired and that the magnitude of the tolerances depends upon the allowable variation of the allowance that is acceptable for the fit.

Since tolerances are unavoidable there will be for each kind of fit and nominal size a *minimum allowance* and a *maximum allowance*. The application of these terms to both loose and tight fits would necessitate the use of the signs or the terms "plus" and "minus" to indicate whether the allowance were a clearance or an interference. It is found desirable in writing or speaking of allowances to avoid the signs or terms "plus" and "minus." Hence the terms "minimum clearance" and "maximum clearance" for loose fits and "minimum interference" and "maximum interference" for tight fits, are to be preferred to the terms "minimum allowance" and "maximum allowance" used positively or negatively. Since the actual size of a machined part will usually be somewhere between the limiting sizes set, the actual allowance of mating parts will be somewhere between the minimum and the maximum clearance or interference specified, and therefore will be nearer the most desired allowance than either the minimum or the maximum clearance or interference. Hence, for any given fit, neither the most preferred allowance nor the allowance attained is either the minimum or the maximum clearance or interference. The allowance has been defined by the A.E.S.C. Committee as the minimum clearance or maximum interference of mating parts: that is, the clearance or the interference corresponding to the largest shaft and smallest hole. There would seem to be a question as to whether it is necessary or desirable to define the minimum clearance as the allowance for loose fits and the maximum interference as the allowance for tight fits. The minimum and maximum clearances are the minimum and maximum allowances for loose fits, and the minimum and maximum interferences are the minimum and maximum allowances for tight fits. "Allowance," as a general term applying to both loose and tight fits, seems too valuable a term to be restricted in any way.

Proper and reasonable clearances and interferences for the various kinds of fits have, for the most part, evolved from practice and experience. Especially is this true of the various kinds of running fits. The theoretical interference, also clearance, for a push fit is zero; hence the theoretical tolerances for the male and female members would be zero. Since tolerances are unavoidable the push fit must be practically defined. The fitting parts may assemble by hand with comparative ease or with difficulty, or light blows may be employed in their assembly. It is, however, evident that to secure a push fit low tolerances, and therefore a high grade of finish, must be used. This is also true of close running fits where the clearances are small. For the usual proportions of hub practice, mathematical and experimental investigations fix the desired interference for the force or shrink fit at about 0.001 in. per inch of diameter, the desired interference for the drive fit naturally falling about midway between the desired interference for the push and for the force fit, or about 0.0005 in. per inch of diameter. Hence, if the limiting sizes of the mating parts are made to differ by too much—that is, if too large tolerances are used—the resulting fit of supposedly interchangeable parts may be anything from a push to a force fit. In the same way different kinds of running fits may meet or overlap. However, for fits having a very narrow range of variation of the clearance or interference, small, expensive tolerances can, to a certain degree, be avoided by sacrificing strict interchangeability and resorting to selective assembly.

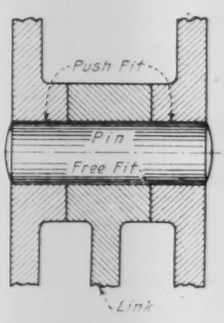

FIG. 1

HOLE VS. SHAFT AS BASIS OF FIT-SYSTEM

Assuming reasonable tolerances and allowances known, a comprehensive system of fits for a desired range of nominal sizes cannot be fully set forth until it is decided whether the system is to be based on the hole or on the shaft. If the hole is taken as the basis, the diameter of the shaft is varied to secure the allowance and kind of fit desired. If the shaft is taken as the basis, the diameter of the hole is varied to secure the allowance and kind of fit desired.

Where the holes to be created are all small, say, not over 3 in. in diameter, the hole as a basis has a decided advantage over the shaft as a basis. This is true where the holes are small since they can be reamed to size. For any given grade of work only one reamer for each nominal size would be needed for all kinds of fits instead of a reamer for each kind of fit for any given nominal size, as would be the case were the shaft the basis. Since mass-production methods are usually applied to small machines having small parts, this advantage of the hole as a basis may be considerable. Similarly, the hole has a decided advantage over the shaft as a basis with respect to the number of mandrels required.

For gaging shafts, "go" and "not go" gages instead of being made

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in one piece are frequently made of four separate measuring fingers attached to a distance piece, the ends of the distance piece being made to the limiting dimensions for the shaft diameter. When the measuring fingers have become worn, the gage may be quite easily restored to its original status by detaching, truing-up, and replacing the measuring fingers. This principle of the division of functions cannot be applied to "go" and "not go" gages for gaging holes. Hence for caliper gages the hole has an advantage over the shaft as a basis of a system of limits, since for the hole as the basis the majority of the gages required are for gaging shafts.

A system based on the hole breaks down, however, where there is more than one kind of fit on the same shaft or spindle, but this may not mean that the hole as a basis should be discarded. If, for the manufactured product, two or more different kinds of fits on the same shaft occur only infrequently, it might easily pay to retain for general use throughout the plant a system based on the hole and to create special holes where more than one fit is required on the same shaft. Thus the hole basis would require for the example in Fig. 1 that the mid-portion of the pin be made 0.9986 in. $^{+0.0000}_{-0.0013}$ and the end portions 1.0000 in. $^{+0.0000}_{-0.0004}$ in diameter, all the holes being 1.0000 in. $^{+0.0006}_{-0.0000}$. Obviously this would not be constructive. The difficulty can be overcome by making the pin 1.0000 in. $^{+0.0000}_{-0.0004}$ in diameter throughout and making a special hole in the link 1.0014 in. $^{+0.0013}_{-0.0000}$ in diameter, the tolerance and minimum clearance being as before.

Where the holes are so large that reamers would not be used, or where conditions are such as not to warrant an investment in reamers for even the majority of holes that could be reamed, the shaft is a better basis for a system than the hole. This might be true even where all of the holes are of such size as to be easily reamed, as where cold-rolled shafting of various sizes is purchased in quantity and used as purchased for male members. The shaft as a basis does not break down and has no exceptions in application. In this respect it is superior to the hole as a basis.

For the reason that in some plants a system based on the hole is decidedly better than a system based on the shaft while in other plants the reverse may be true, the author favors the German practice of having two standard limit systems, one based on the hole and the other based on the shaft.

Best Method of Expressing Tolerances

Having fixed upon suitable tolerances and allowances and having fixed upon the hole or the shaft as the basis, it remains to be decided how best to express the tolerances, whether *unilaterally* or *bilaterally*. About this matter there has been considerable discussion, and the adoption of a standard system of fits awaits a settlement of whether the tolerances shall be expressed unilaterally or bilaterally.

The author was brought up to express tolerances bilaterally, and it appears to him that the mistake has been made of claiming advantages for the unilateral over the bilateral method that do not exist, thereby alienating individuals having bilateral prejudices. The advantages of the unilateral over the bilateral method are few but real, and are of sufficient importance to be decisive. So far as accuracy of size is concerned, the decisive advantages appear to be that the unilateral is more convenient to use than the bilateral method of expressing tolerances and may require fewer gages.

The designer and the consumer are interested in having mating parts fit and function properly; that is, they are interested in being assured of acceptable allowances. The designer is also interested with the producer in securing acceptable allowances at a minimum cost; that is, the designer and producer are interested in tolerances and the most convenient and economical method of expressing them. Whether the designer determines a size rationally or empirically, it is rounded to some convenient nominal size. No matter what the fit or how the limiting sizes may be determined and recorded on the drawing, he is assured that the actual sizes of the mating parts will be sufficiently close to the nominal to answer every purpose. Hence, with respect to accuracy of size for fits, the nominal size is not a desired, ideal size, and the designer is free to express the tolerances in any way that would prove most convenient and economical. If the use of the unilateral method is in no way more expensive and is, in at least one decisive way, less expensive, it should be favored over the use of the bilateral method.

TABLE 2 COMPARISON OF FITS FROM THE NEWALL AND PROPOSED A.E.S.C. SYSTEMS

Kind of fit	For 2 in. nominal size — Tolerance — Hole	Shaft	Clearance or interference Min.	Max.
Newall easy fit work (Class X)	0.00150	0.00175	+0.00125	+0.00450
A.E.S.C. medium fit (No. 3)	0.00100	0.00100	+0.00140	+0.00340
Newall push fit (Class P)	0.00100	0.00050	0	+0.00150
A.E.S.C. wringing fit (No. 5)	0.00075	0.00050	0	+0.00125
Newall force or shrink fit (Class F)	0.00100	0.00100	−0.00225	−0.00425
A.E.S.C. heavy force or shrink fit (No. 8)	0.00075	0.00075	−0.00200	−0.00350

Comparison of Newall and A.E.S.C. Systems

Three fits from the Newall system and from the system proposed by the Sectional Committee of the A.E.S.C. are contrasted in Table 2. These fits are comparable since in each case the minimum or maximum clearances or interferences are about the same, and therefore the fit is the same. Each system is based on the hole. The Newall system uses two grades of holes while the A.E.S.C. uses four, a different grade of hole for each of the three running fits named in Table 1 and another grade of hole for the remaining five fits. Hence, the A.E.S.C. system might require, for the manufacture of a given product, more reamers and mandrels than the Newall system. For any given nominal size two reamers and two mandrels would be required for all of the fits under the Newall system, while for all of the fits under the A.E.S.C. system four reamers and four mandrels would be required. However, it must be remembered in contrasting the two systems that the work of any given plant may not require all three classes of running fits, especially for the same nominal sizes.

In Table 3 the tolerances for the three A.E.S.C. fits of Table 2 for a nominal size of 2 in. are expressed bilaterally and unilaterally, using the hole as the basis of the system. The *basic size* of a part has been defined as the size from which variation is made to secure the limiting sizes. The basic size of the hole has been taken equal to the nominal size. In applying the bilateral method to the basic hole one-half of the tolerance has been added and one-half subtracted from the basic size; in applying the unilateral method the entire tolerance has been added and zero subtracted from the basic size. In applying the bilateral method to the shafts, the basic sizes have been taken such that the addition and subtraction of one-half the tolerance from the basic size of each shaft yields the limiting sizes; in applying the unilateral method the basic sizes have been taken such that the addition of zero and the subtraction of the tolerance from the basic size of each shaft yields the limits necessitated by the minimum and maximum clearance or interference.

TABLE 3 TOLERANCES FOR THE A.E.S.C. FITS OF TABLE 2 EXPRESSED BILATERALLY AND UNILATERALLY, USING THE HOLE AS THE BASIS OF THE SYSTEM

(Nominal size, 2 in.)

	Medium fit	Wringing fit	Force or shrink fit
Bilateral Method:			
Dimensions of hole	2.0000 ±0.0005	2.0000 ±0.000375	2.0000 ±0.000375
Dimensions of shaft	1.9976 ±0.0005	1.999375 ±0.00025	2.00275 ±0.000375
Unilateral Method:	+0.0010	+0.00075	+0.00075
Dimensions of hole	2.0000 −0.0000	2.0000 −0.00000	2.0000 −0.00000
Dimensions of shaft	1.9986 +0.0000 −0.0010	2.0000 +0.0000 −0.0005	2.0035 +0.00000 −0.00075

With respect to the use of adjustable and double-ended limit gages, there is no advantage, for the fits of Table 3, of either method over the other of expressing the tolerances. If, however, separate "go" and "not go" gages are assumed, it is evident from inspection of Table 3 that the dimensions for the three fits with the tolerances expressed bilaterally would require 3 "go" and 3 "not go" gages for the shafts and 2 "go" and 2 "not go" for the holes, or a total of 10 gages; with the tolerances expressed unilaterally 3 "go" and 3 "not go" gages are required for the shafts and 1 "go" and 2 "not go" for the holes, or a total of 9 gages. Hence by expressing the tolerances unilaterally a saving of one gage may be effected for the three fits. It is also to be noted that with respect to fits the unilateral method is more convenient to use than the bilateral.

Standard Method of Expressing Tolerances Much Needed

The adoption of standard tolerances and allowances for different

kinds of fits with a standard method of expressing the tolerances is very much needed. Such a standard system, made generally available, would lead to better design, to increased production, and to a decrease in the cost of production. It would furnish many designers a better basis for the exercise of judgment than they now have, not only with respect to cylindrical fits but with respect to longitudinal fits and the fit of guides and slides.

All that has been said above is with respect to accuracy of size, with accuracy of form assumed. With respect to accuracy of position, which is often of more importance than the accuracy of size for fits, the bilateral method may, in certain cases, be better than the unilateral method of expressing tolerances.

If it is desired to place a tolerance on the distance between the shafts on which two involute gears are mounted, it would, in general, be better to express the tolerance unilaterally because of the properties of involute teeth. Such teeth will transmit motion equivalent to the pure rolling of the pitch circles whether the distance between shafts is over or under the desired nominal distance of, say, 10 in. If, however, the gears are accurate as to size and form, a decrease in the distance between centers would be objectionable while a certain increase would not be. Hence, it would be logical to express the distance between centers as 10 in. $^{+0.005}_{-0.000}$ It is to be noted that the tolerance is expressed unilaterally and as a variation from a desired, or *ideal*, dimension. If the grade of work were such that 0.001 in. under and 0.004 in. over would be reasonable, it would be best to express the distance as 10 in. $^{+0.004}_{-0.001}$ rather than as 9.999 in. $^{+0.005}_{-0.000}$; that is, bilaterally rather than unilaterally, clearly showing that a greater increase than a decrease from the ideal dimension is permitted. If, in addition, it is desired to specify the degree of parallelism of the shafts, the permissible variation would be thought of as applying either way so long as the distances between the supporting bearings at the ends of the shafts were within the limits previously set.

In the case of two cycloidal gears a variation from the ideal center distance is as objectionable in one direction as in the other. In order that the desired, or *ideal*, dimension may appear 10 in. ± 0.002 is to be preferred to 9.998 in. $^{+0.004}_{-0.000}$. In the spacing of gear teeth for service requirements that are exacting, the permissible variation is thought of as bilateral rather than as unilateral. If a variation of spacing of 0.0001 in. is to be tolerated, it is thought of as applying either way from the true spacing, an acceptable space being either smaller or larger than the ideal.

Where accuracy of position depends on right angularity, it is often possible to express the permissible variation as a small fraction of an inch in a certain length, the variation being assumed to apply either way from a true right angle.

It is also true that when, to secure accuracy of position, the dimensioning is from datum lines to centers and shoulders and surfaces, bilateral variation is often the basis for determining the limiting distances. On the finished drawing it may be found best in some cases to express the permissible variations bilaterally and in other cases unilaterally. The case is different than with fits. For a 2-in. fit of any kind no one cares, so long as the parts fit properly, whether either mating part is exactly 2 in. The case of accuracy of position is often different in that there is a desired, *ideal* dimension for which we are striving.

Measuring Systems and Tolerance Limits

Fit-Desirability Diagrams and What They Show about Systems of Measuring

By P. J. DARLINGTON,[1] BOSTON, MASS.

THE DESIGN of machines by the application of mechanical laws and knowledge of the properties of the materials has developed into an almost exact science up to the determination of cylindrical fits, or the clearances and forcing allowances between holes and their shafts or seats. The remaining technical problem of mechanical industry appears to be that of precision, or the value-versus-cost determination of tolerance limits and their application to production.

Little is known of the relation of precision or good workmanship to the performance of machinery. Industry is not well situated to gain this from experience, because whatever may have happened from a tolerated inaccuracy is the result of an unrecorded cause in that individual machine. Also, the evidence of the parts themselves is usually destroyed when they fail, and in the laboratory we are handicapped by the inability to reproduce all of the practical working conditions.

The published tabulations of tolerance limits representing average practice are interesting chiefly in showing the very large tolerances employed in proportion to the fit allowances, or the large sacrifice of factor of safety in present practice, which emphasizes the practical importance of the subject.

While science and industry are adding to the data available, there appears to be no scientific basis for the employment of what data we now have and may hereafter acquire. Tolerance limits and measuring systems are, in fact, very little understood and very inefficiently employed in many factories, in some of which the entire scheme of precision control is in danger of collapse because of the evident damage to product and increased cost resulting from misuse of tolerance limits and the employment of gaging and measuring means unsuitable to the operations on which they are used.

It is the object of this paper to put this subject of tolerance and measurement on a scientific basis by developing and discussing a new method of employing efficiently all that we have, or may acquire, of data, experience, and personal judgment in the setting and use of tolerance limits.

A recent experience of the author's may serve to illustrate the very practical bearing of this subject on the performance of machinery. In an automobile, towed in for repairs to the rear end, a hard steel sleeve had worked loose from its seat on the transmission shaft, thereby destroying the shaft, sleeve, and bearing. The repair man drew replacement parts from stock and proceeded to place the new sleeve on the new shaft. After much hard work the sleeve was stuck fast, about half-way on. He discarded that sleeve and shaft, taking another pair from stock. The sleeve was finally seated, but the parts had been so abused that the final success of the job was in doubt. The failure in service appears to have resulted from too loose a fit of the original parts, while the difficulties of repairing were due to the opposite defect, or too tight fits of the replacement parts. It is probable that the correct diameters were specified in manufacture and that the failures were due to too wide tolerances of production.

Fit-Desirability Diagrams

In view of all considerations of service and security there is, of course, always one best fit, which we assume to be the difference of the diameters of shaft and hole calculated by the engineers and specified on the drawings. This we may call the 100 per cent desirable fit. Any other fit, either tighter or looser, would be less desirable. The quality of that part would be impaired. The part could be used only at some risk of trouble or expense in service or repairs.

If we could value in terms of fit allowance all the elements entering into desirability and risk, such as smooth operation, power, efficiency, durability, reliability, and repairability, we might plot a diagram or curve such as that shown in Fig. 1 (not actual) in which—

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x = fit, or diameter of shaft less diameter of hole
y = per cent of available desirability, or safety factor
r = $100 - y$ = per cent avoidable impairment or risk
d = fit specified by the engineers on the drawings.

In some cases this curve might be plotted from available data. For example, suppose that we are to produce shafts, without keys,

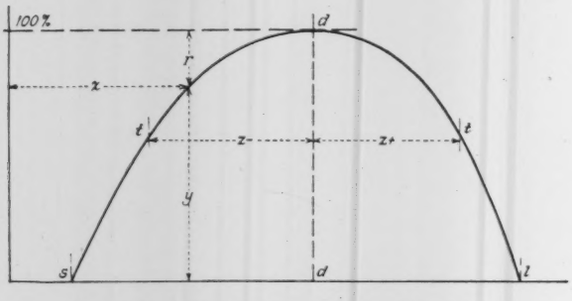

FIG. 1 TYPICAL FIT-DESIRABILITY DIAGRAM

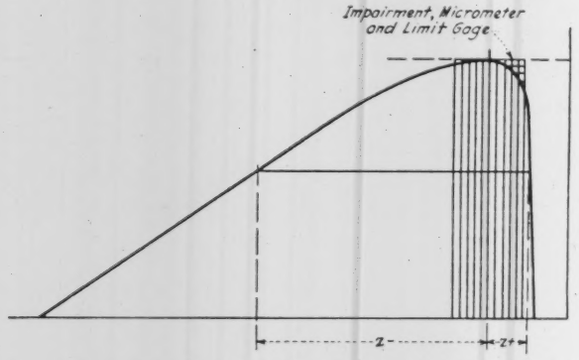

FIG. 2 FIT-DESIRABILITY DIAGRAM WITH SMALL PLUS AND LARGE MINUS TOLERANCES

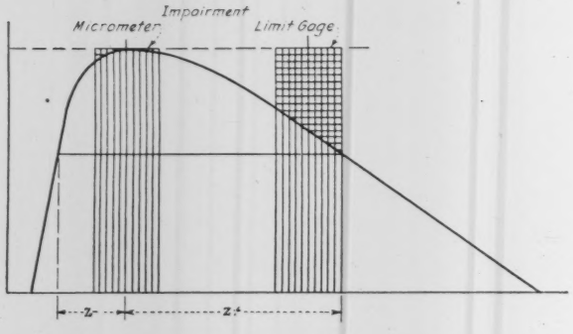

FIG. 3 FIT-DESIRABILITY DIAGRAM WITH LARGE PLUS AND SMALL MINUS TOLERANCES

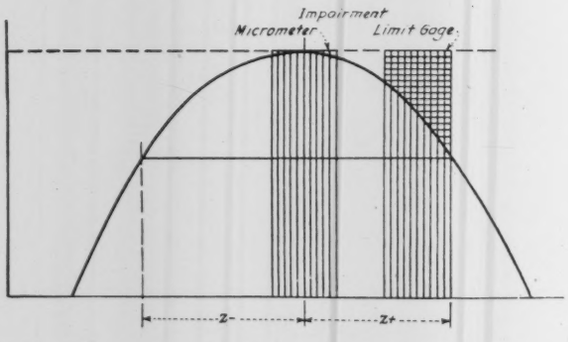

FIG. 4 FIT-DESIRABILITY DIAGRAM WITH EQUAL PLUS AND MINUS TOLERANCES

to be forced into the hubs of motors to transmit a known torque in service, and yet to be removable with a hydraulic jack of known capacity in case of need for repairs.

From force-fit tables the two values of x for each assumed factor of safety y, which we will consider the measure of desirability, might be determined. At $y = 0$, $x = s$, or that prohibitively loose fit which will just fail to carry the working torque of the motor, and $x = l$, or that prohibitively tight fit at which the hydraulic jack would just fail to start out the shaft for repairs.

Having once determined the form of the characteristic curve for such forcing fits, we might apply it without calculating intermediate points to fill in between s, d, and l in similar cases. We might, with the help of force-fit tables, vary the scale of x or y to make the curve applicable to different diameters.

It is but seldom, however, that the elements of desirability are so simple and measurable, or that so many data are available. This force-fit example has been selected in order to explain the nature of the fit-desirability diagram.

Fig. 2 (not actual) might be the characteristic diagram for the fit of nuts on bolts. It is desirable to have the nuts as snug as may be, but free enough to spin on to the bolts with the thumb and finger. A little looseness may not do much harm, but if they are too tight the assembler will have to use a wrench to run them all the way up on the bolts, which is prohibitive. This results in an unsymmetrical diagram.

Fig. 3 (not actual) might be the fit-desirability diagram for grinding shafts to a specified diameter for subsequent lapping to a finish size. In this case the finish for lapping replaces the fit allowance, but the problem is the same. We want to grind close to size and we have substantial tolerance over, but very little under, size. It is the reverse of the conditions of Fig. 2.

A symmetrical diagram, with equal plus and minus tolerances, approximated in Figs. 1 and 4, would apply to the production of commercial drill rod or other articles when we do not know what they will fit with or how they will be used.

In the case of running fits the elements of desirability are many and we have, as yet, very little exact knowledge of the relation of clearance or oil film to efficiency, durability, and reliability in actual service.

DETERMINATION OF TOLERANCES

We always know the point at the top of the curve $y = 100$, and $x = d$ or drawing size. We can usually determine approximately the prohibitively loose and tight fits s and l, and we can sketch in the curve between these three points. If we had no knowledge whatever of the effect of fit on desirability, and of neither s nor l approximately, we could not make any diagram, for tolerances would then be of no significance; they would be mere arbitrary instructions to workmen and inspectors.

It will be easily understood how the fit-desirability diagram may be used to determine plus and minus tolerances, $z+$ and $z-$, for any total tolerance $t - t$ and corresponding impairment of product or risk r, all as agreed upon in consideration of the need of precision in that operation versus the shop cost of producing it. The tolerances $z+$ and $z-$ are of course to be divided between hole and shaft in the most economical proportions for the two operations, remembering to reverse the plus and minus, as enlarging the hole is equivalent to reducing the shaft.

The sketching in of the diagram between the three given points has of course an element of guesswork, which can be reduced only as science and industry furnish additional data to replace it. The solution is not, therefore, as yet entirely quantitative, and it may be unsatisfactory to the mathematically minded student. The form of this diagram may, however, be a means of graphically expressing, comparing, and discussing the judgment of individuals as to the effect of varying fits. It may eventually become a basis of some kind of standardization for the industries.

FACTS BROUGHT OUT BY THE DIAGRAMS

In the meantime this method of treatment brings out very clearly some important general principles having practical bearings on production and performance, and perhaps explaining some puzzling shop experiences.

1 From the general nature of the elements entering into desirability and the balancing of opposing risk elements at $x = d$, we may reasonably assume that, in general, the fit-desirability diagram is round-topped or convex upward. Therefore dy/dx increases in both directions from $x = d$, which means that as we gradu-

ally widen tolerances and permit greater inaccuracy of production, the impairment r of the product increases faster, perhaps much faster, than in constant proportion.

2 It follows that it may be very important, and often urgent, that we reduce limits reasonably, while it may not be worth the cost to reduce them too far. It is the outer edge of the tolerance zone that does the damage.

3 It is clearly essential that we first work out the total tolerance for the fit and then divide it between shaft and hole, because the total risk may be far greater than the sum of the risks. Shaft and hole tolerances, worked out separately, may each look more than doubly safe, and yet result in misfit when the extremes meet at assembly.

4 Limit gages are made to the extreme tolerated sizes and they therefore work on the outer edge of the tolerance zone where an error in one direction causes maximum impairment of product. Wear of the "go" plug gage and wear and spring or crowding of the "go" snap gage are all in the harmful direction, or such as to bring the diameters out of tolerance bounds. Moreover, and unfortunately, the wear of the plug gage is in the same direction on the diagram as that of the snap gage, and these add on the same side of the diagram to produce an unauthorized extension of tolerance limits on that side, and at a place on the slope of the diagram where the impairment of product r is greatest. The damage done by the combined error may be, as we have seen, much more than the sum of the two separate effects. We therefore conclude that inaccuracy with a limit gage may be far more serious than an equal inaccuracy with a measuring instrument such as a micrometer which aims at the correct size d and works at the top of the fit-desirability curve.

5 We may also conclude that a worn plug gage may be very bad and a worn limit snap gage also very bad, but both together on the same job will be very much worse than twice as bad.

6 Tolerance limits are not a true indication of precision even in measurement, for one skilful workman with a micrometer may keep close to size and not use all the tolerance, and another less skilful may scatter his sizes within the tolerance zone, while a man with a limit gage, no matter how skilful, will naturally bunch his sizes toward the "go" side of the tolerance zone.

In Figs. 2, 3, and 4 such bunching of the sizes produced is illustrated by the shaded zone, and the total impairment of product by the cross-hatched area. To what extent this bunching of sizes actually occurs in practice with different measuring, gaging, and dimensioning systems might be the subject of a very useful investigation. Fig. 2 may illustrate the reason for the advocacy of the so-called "unilateral" tolerance system by bolt and screw manufacturers.

We conclude that one cannot judge the precision quality of product by the tolerance limits to which it was manufactured, especially without due consideration of the measuring system employed and the suitability of that system to the case in hand as shown by its fit-desirability diagram.

7 From a comparison of the cross-hatched areas of Figs. 2, 3 and 4 showing impairment of product, we may conclude that limit gages and drawings dimensioned only in limits may produce good product in the special case of Fig. 2, such as bodies and threads of bolts and nuts, in which there is extremely small plus tolerance. In all other cases such as forcing and running fits and especially in reducing for a subsequent finish operation, as in Fig. 3, the limit-gage system is unsuitable.

8 We may also conclude that limit-gage systems are especially dangerous in factories where tolerances are not well understood or not carefully computed and may be excessive on the plus side (plus for shaft). A micrometer aiming at a correct diameter need not employ the excessive tolerance, which becomes useless but harmless. Limit gages, however, have the excessive tolerances built into them and must produce corresponding oversize product—not occasionally, but all the time.

9 If a micrometer be correctly adjusted or set to zero the inaccuracies of using it will be only occasional and at the top of the fit-desirability diagram where r is small and but little impairment of product results. On the other hand, any fixed or constant error of the instrument results in shifting the target, and moves the average of all readings into a position of greater impairment while passing occasional sizes beyond one tolerance limit. The result, however, is not only increased general impairment but also risk of rejections.

We therefore conclude that errors of applying and reading a micrometer are far less harmful than equal errors due to lack of adjustment for wear, to the personal factor of the user, or to expansion from handling.

Micrometers and other measuring instruments should therefore be checked and adjusted frequently by the user himself on a cylindrical standard of about the diameter of the work, and preferably mounted to avoid handling and expansion of the standard. In short, a micrometer should be employed as a comparator, and time may be better spent in frequent checkings and adjustment than in exercising great care in using the instrument on the work.

10 The total tolerance of the fit is to be divided between shaft and hole, and a unit of tolerance has equal effect in either. The operations of sizing holes are very different from those of sizing shafts, and the cost of precision to any specified tolerance will usually be much greater on one than on the other. Failure to apportion correctly the available tolerance between hole and shaft may lead to greatly increased cost, and is probably the source of great loss in industry. An investigation of the cost of precision in different operations might therefore be expected to lead to a tabulation of data of the greatest practical value.

11 From all the foregoing we may conclude that in operations of production the highest degree of actual precision can be obtained only with drawings which specify, and measuring means which aim at, the exact most desirable diameter. The most efficient use of tolerance limits is in a scientific division into plus and minus and a scientific sharing of each between hole and shaft. Frequent checking and adjustment of the measuring instrument on the standard is more important than is generally known.

Limit gages appear to be more applicable to final inspection than to production.

It is difficult for a workman with a hand micrometer to use drawings giving the three dimensions of size and unequal plus and minus tolerances, especially if all are fractional. He must translate their numerical values on the drawings to readings on the micrometer scales and carry all three in mind.

We may therefore expect the rapid development and adoption of production measuring instruments employing the comparator principle for eliminating errors and having mechanical or graphic means of displaying the plus and minus tolerance limits for guidance of the user.

The Vickcen Centrifugal Oil Separator

DESCRIPTION of a device made by the British Separators, Ltd., London, for which it is claimed that the separating action adjusts itself automatically to the proportions of pure and impure liquids. Thus, if pure liquid were fed into the bowl there would be no discharge of impure liquid already in the bowl, the only discharge being that of pure liquid from its proper outlet. Similarly, if liquid impurities only were fed in, there would be no discharge of the pure liquid in the bowl.

The particular features of the device lie in the design of the bowl, which is equipped with a number of conical plates separated by distance ribs and provided with holes formed at intervals round the periphery of each plate at top and bottom. The liquid passes through these holes and under the influence of centrifugal force spreads itself in a thin layer on the inside of each conical plate. The application of centrifugal force to a number of thin films of liquid instead of to the liquid in a mass is an important principle in the design of the separator, it being maintained that with the liquid disposed in this manner, separation is instantaneous and complete.

For high clarifying of the liquids which have only a slight proportion of impurities, the bowl arrangement is modified so that there is no outlet for impurities, the whole of these being retained in the sediment space.

The original article describes other details of the design and methods of drive. (*Mechanical World*, vol. 78, no. 2018, Sept. 4, 1925, pp. 179–180, 4 figs., *d*)

The Prime Cause of Friction

Cohesion, under Certain Conditions, Friction—Friction Wear Merely Minute and Continued Breakage

By ERNEST W. DAVIS,[1] OAK PARK, ILL.

THE usual definitions of friction define this force as some specific resistance to motion. They often describe certain conditions under which friction may occur, but, owing possibly to the apparent absence of any hypothesis explaining the cause of friction, do not consider in any way the nature of friction itself. In the hypothesis hereinafter advanced, it is assumed that friction is always caused by the resistance to separation of molecules under mutual attraction, and that friction cannot be overcome without causing the separation of such mutually attracted molecules.

Cohesion, under Certain Conditions, Friction

When considering the nature of friction it is necessary to keep in mind the fact that there is no such thing as a perfectly smooth surface. Surfaces which appear to be perfectly smooth, if magnified sufficiently, will reveal numerous small hills and valleys, so that when two such surfaces are brought together there is more or less interlocking of the projecting portions. Bearing this in mind, let us suppose that we have two pulleys fastened to a shaft, one a split pulley with the two halves clamped together tightly so as to grip the shaft, and the other a pulley keyed to the shaft. In the split pulley the minute projections on the shaft and in the bore of the pulley interlock, and the resulting friction of rest tends to prevent rotation of the pulley relative to the shaft. In the keyed pulley the function of the interlocking projection is performed by the key. Although one pulley is held by friction and the other by a key, the two conditions are quite similar. The rotation of either pulley with respect to the shaft must shear off some interlocking projection, and in either pulley it is cohesion of the metal in the interlocking portions that tends to prevent rotation. In other words, the conditions tending to prevent rotation of the two pulleys are fundamentally the same, the difference being one of degree only. Each pulley is prevented from rotating by the force of cohesion; the cohesion in one case being located in a large number of small interlocking portions, and in the other case in a single, large interlocking portion. So it would appear that cohesion, under certain conditions, may be called friction.

Other Forces Producing Friction

But cohesion is not the only force that can produce friction. Frictional effects can also be produced by the action of adhesion, and it is sometimes difficult to decide whether the resisting force is adhesion or friction. As an illustration, imagine a smooth board, elevated at one end so as to form an inclined plane, on which a small, smooth block of wood may slide freely from the upper end to the lower under the action of gravity. If the board is inclined sufficiently, the movement of the wooden block may be made fairly rapid. If a coating of hot glue is then applied to the board and the experiment repeated, it will be found that the block descends less rapidly, and by making repeated trials it will be found that the motion becomes slower and slower as the glue loses heat and becomes more viscous. This added resistance to the movement of the block (fluid friction) soon increases to such an extent that the block comes to rest and shortly afterward requires an appreciable force to move it.

If the glue-coated board with the block resting upon it is allowed to stand undisturbed until the following day, the pressure of the hand is no longer sufficient to move the block. The force which resists movement of the block has increased many times, and is no longer known as fluid friction but as adhesion. Here we have a condition in which a force changes its name, so to speak, by merely increasing in magnitude. Although the increase in magnitude occurs by imperceptible degrees, there seems to be no intermediate stage between fluid friction and adhesion. The question is, at what precise stage in the gradual hardening of the glue does friction change to adhesion?

If sufficient pressure is applied to the wooden block to overcome the adhesion of the hardened glue, the glued joint will be broken and the block will move. It is important to notice that something breaks. This breakage, it would appear, is a determining factor in differentiating between friction and adhesion. When the resistance due to the action of plastic glue is overcome, no breakage occurs, and the resistance is called friction; but when the resistance due to the action of hard glue is overcome, then there is breakage of the glued joint and the resistance is called adhesion. Referring again to the split and keyed pulleys, we find that there, too, breakage is a determining factor in differentiating between cohesion and friction. If sufficient force is applied to rotate the split pulley on the shaft, breakage does not occur—at least no perceptible breakage occurs—and the resisting force is called friction; but if sufficient force is applied to rotate the keyed pulley, the key will be broken, and the resisting force is then called cohesion, although the two forces are fundamentally alike.

Wear Due to Friction a Minute and Continual Form of Breakage

If friction and cohesion are fundamentally the same, then when we overcome either friction or cohesion we should expect to produce effects that are fundamentally the same. And we do. It is true that overcoming friction produces wear while overcoming cohesion causes breakage, but there is really no fundamental difference between wear and breakage. The difference is one of degree only. If a perceptible portion is forcibly separated from a body we say that it is broken off, but if many imperceptible portions are separated we call it wear. *Wear is merely minute and continued breakage.* If we strike a rock with a hammer the rock may be broken. If we rub the rock against another it will be worn. In either operation the force of cohesion tending to hold the particles of rock together must be overcome, and in either operation force is required to overcome this cohesion.

Wear Always Results When One Surface Slides Over Another with Which It Is in Contact

When one surface slides over another with which it is in contact there is always some wear. A single tick of a watch produces wear in the moving parts. We may not be able to detect such a slight amount of wear, but we can detect the wear produced by a billion such ticks, so that we can reasonably assume that the wear of one tick is approximately one-billionth as much. During the wear which always occurs when contacting surfaces rub against each other, myriads of mutually attracting molecules are separated from each other. Each molecule thus separated offers resistance to its separation, and this accumulated resistance is what we call friction.

It may be argued that in the case of sliding friction the irregularities of the two rubbing surfaces merely slide over each other and in that way produce resistance without being actually displaced. If this were true then we could slide one surface over another without causing wear—something that has probably never been done. It is no doubt true that many of the larger irregularities actually do slide over each other without being removed; but if we consider that the sides or faces of such irregularities are covered with smaller irregularities, and that the faces of these smaller irregularities may have still smaller irregularities, and so on, then it is difficult to conceive of motion between two solid surfaces in contact with each other without actual, permanent displacement of some of the molecules.

The author hopes that the foregoing hypothesis may stimulate thought in a new direction, and that a constructive criticism of the views he has expressed will follow, leading ultimately to further developments in this field, and possibly helping to overcome the difficulties encountered in accurately formulating any fundamental law of friction which will apply under the wide range of conditions experienced in engineering practice.

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An Instrument for Recording Vibrations

By L. H. YOUNG,[1] CAMBRIDGE, MASS.

A PORTABLE instrument for measuring and recording the amplitude and frequency of vibrations encountered in buildings, bridges, and small earth tremors has recently been developed in the Laboratory of Industrial Physics at the Massachusetts Institute of Technology. Although primarily intended for the civil engineering profession, mechanical engineers will find the instrument of advantage in investigating the vibrations of machinery, engines, turbines, etc., failures of machine tools due to vibrations, and many similar problems which will suggest themselves to the reader.

Vibration is the cause of much difficulty in industry, and because of the lack of quantitative data, little has been done to attack this problem from a logical standpoint. By experience the structural engineer has found that certain types of structure are suitable for the housing of particular types of machinery, and this knowledge has often been gained at too great an expense. Serious difficulties arise when two dissimilar industries occupy the same structure, one giving rise to vibrations of such frequency and amplitude as to seriously impair the production of the second. These cases are numerous and various remedies have been applied. The stiffening of the structure and relocation of machinery have often solved the major part of the difficulty. A systematic study of such cases would give the frequency, amplitude, and plane of maximum vibration of the structure, and with this knowledge at hand a logical basis could be obtained upon which to plan the necessary changes. A comparative study should be made of various buildings and of information as to their type of construction, the machinery they contain and its location, and such other particulars as seem to bear on the case. A study of a large amount of such information would undoubtedly yield data upon which plans could be made to avoid future difficulties. Such comparative data for buildings or bridges on the effect of ground tremors caused by traffic or blasting, if systematically studied, would indicate the desirability of particular types of foundations in order to avoid vibration.

From a study of numerous seismograph references the author has arrived at the conclusion that these instruments in their present form are unsuited for the purpose in view of their cumbersomeness. They are also not adapted to measure the frequencies and amplitudes encountered in structures. However, the fundamental basis for measurement utilized in seismographs, the "steady point," has been applied in the development of a vibration recorder. A weight so suspended that its position remains fixed in space regardless of the movement of its surroundings is termed a "steady point." A pendulum of comparatively long period in comparison to the forced period to be measured approximates this definition. A proper linkage system between the steady point and surroundings will serve as a means to record the relative motion.

Figs. 1 and 2 show the general arrangement of the instrument for measuring horizontal displacements. Light from the flashlight bulb E is allowed to fall upon a small fixed mirror F. It is reflected through a lens L to a second mirror R. After this second reflection at R it is brought to a focus at the S, where its motion is recorded on a moving photographic film. The mirror R is mounted on an axis, the pointer B bearing on a short lever arm attached to this mirror. The steady-point arm P is connected to the instrument proper at A by two strips of phosphor bronze. The magnification of the instrument may be readily varied between 20 and 300 by moving the unit L, R. The metal vane D, which is enclosed in a housing to prevent transient effects due to air currents, serves as a damping device. The steady point and its arm form a horizontal pendulum. For stability the weight W must be slightly below the supporting strips. This pendulum is simultaneously affected by horizontal motion, rotation about a vertical axis, and effects due to tilting, the latter being due to a change in stability. In general, where horizontal displacements are large the verticals are small, and the effects due to rotation and tilting are also small. By the addition of a suitable spring mounted in a vertical plane, its upper end being attached to a rigid support and its lower end to the steady-point arm, an instrument for measuring vibrations in a vertical plane is obtained.

A Norton accelerometer[1,2,3] was modified by replacing its cantilever spring and dashpot with the steady-point arm and damping vane as described above. The motor drive and optical systems are identical with those of the Norton accelerometer. The details of the motor drive are shown in Fig. 2. The make-and-break

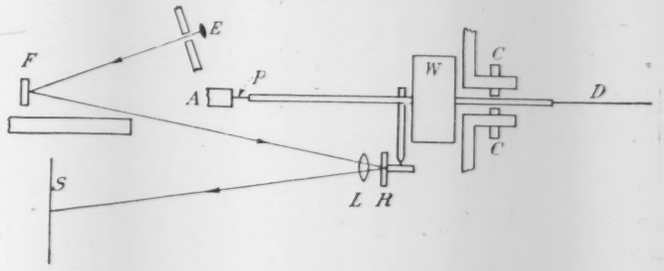

FIG. 1 DIAGRAMMATIC PLAN OF INSTRUMENT FOR RECORDING VIBRATIONS

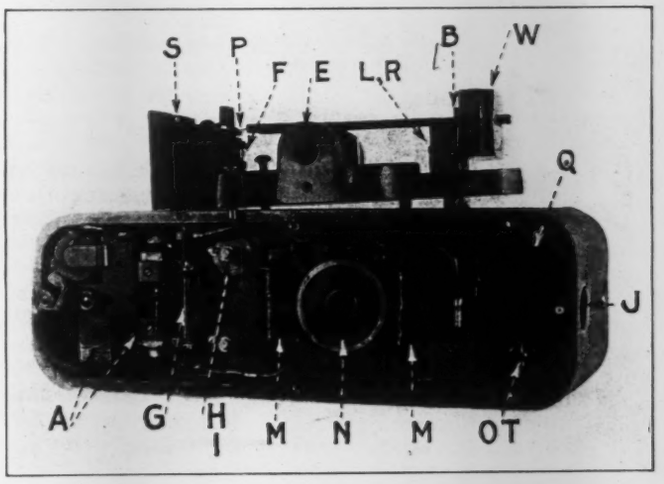

FIG. 2 INSTRUMENT FOR RECORDING VIBRATIONS, SHOWING DETAILS OF MOTOR DRIVE

mechanism A alternately allows coils M to be energized, which in turn attract the rocker carrying two pawls that operate the ratchet wheel N. The latter is geared directly to a large spool around which the film passes and to the winding spool at O, which latter is equipped with a friction clutch to insure a constant film speed of 7 in. per min., regardless of the size of the film roll. As all motor parts move in a plane at right angles to the relative plane of motion of the steady point, the vibration of the motor causes a movement of the spot of light of only 0.01 in. when magnifying 200 times. Two sockets at H and I contain the light sources. The second source is controlled by an external switch, its operation giving a band of light across the entire film width and serving to mark any particular event. The instrument occupies a space 9 in. by 3 in. by 3 in. and weighs 5 lb. Four dry cells are sufficient to operate the motor and light sources.

Numerous records under widely varying conditions have been made to test the adaptability of the instrument. Figs. 3–6 have all been made from prints of untouched negatives.

Fig. 3 is a record of the vibrations of a bench-drill table. The peaks show a frequency of 360 vibrations per minute. A speed-counter check of this showed that the idler-pulley speed was 360

[1] Department of Physics, Massachusetts Institute of Technology.

[1] Jl. S.A.E. vol. 14, no. 2, p. 136, Feb., 1924.

[2] Nat. Advisory Comm. for Aeronautics, Report No. 100.

[3] Jl. Franklin Institute, Aug., 1924.

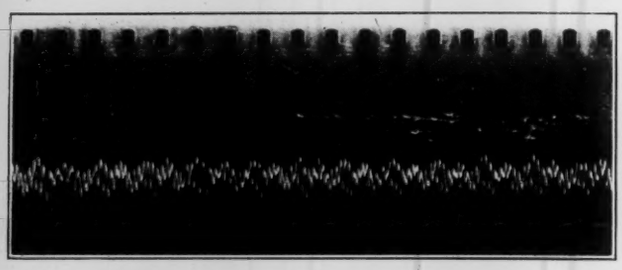

FIG. 3 VIBRATIONS OF A BENCH-DRILL TABLE
(Idler pulley alone rotating.)

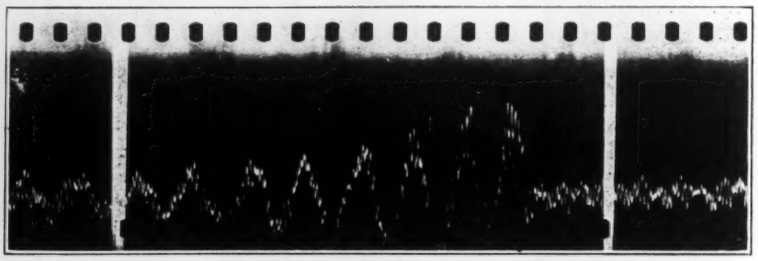

FIG. 4 VIBRATIONS OF DRILL WITH POWER ON OR OFF
(Time between bands, 6 sec.)

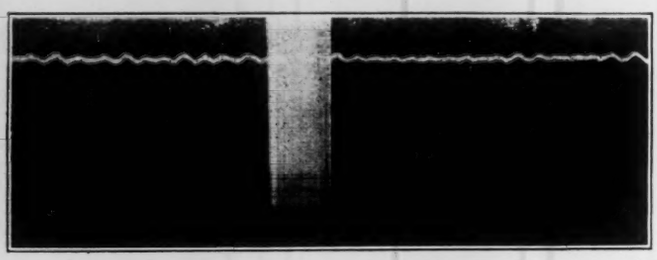

FIG. 5 VIBRATIONS OF FLOOR IN MACHINE SHOP
(Left: Vibrations due to both motor drive and grinding wheel; Right: Vibrations due to motor drive alone.)

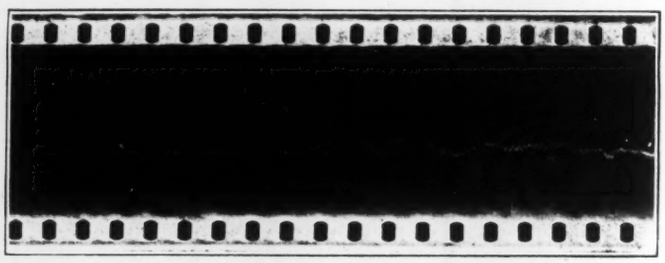

FIG. 6 VIBRATIONS DUE TO PASSAGE OF TRUCK NEAR BUILDING

r.p.m. It was also discovered by a separate timing that the higher peaks occurring 36 times per minute were due to the passage of the belt lacing over the pulley. The record as taken was magnified 108 times, indicating a maximum horizontal movement of 0.0017 in., and was made with the idler pulley of the drill rotating, the driving pulley being idle.

Fig. 4 shows the effect of applying power to rotate the drill and shutting off this power 6 sec. after its application. The maximum displacement is 0.0070 in., the peaks due to speed of rotation being superimposed upon those due to the increased vibration. Part of this large displacement is due to the push and pull on the starting lever of the machine in applying and shutting off the power. The exposed portion above the perforations of the film serves to indicate the constancy of motor speed. Any speed irregularity is here indicated by a variation in the exposure of this strip. Vibration records of this table have been made a number of times with identical results as regards frequency and displacements when different magnifications were used.

The magnification being changed to 210, a record was made of the vibrations of the floor of the machine shop in the basement of the Institute buildings, Fig. 5. The maximum displacement is 0.00027 in. The instrument was mounted near the motor drive and a grinding wheel. The section to the left of the wide band shows vibrations due to both motor drive and grinding wheel, and that at the right those due to motor drive alone.

The right-hand portion of Fig. 6 shows the vibrations due to the passage of a single loaded truck at approximately 30 miles per hour. This record was taken on the first floor of a frame dwelling, and the truck causing the vibrations was moving on the highway at a distance from the instrument of about fifty feet.

The instrument described may also be readily used for recording the vibrations of ships resulting from the unbalance in engines and from propeller action; and the vibrations of individual machines, particularly those of automobile and airplane motors, may be studied with facility by the use of both horizontal- and vertical-type recorders.

Caution in Using Compressed-Gas Cylinders

THE Interstate Commerce Commission has prescribed certain regulations pertaining to the shipment of dangerous articles by rail. These regulations include specifications for the manufacture and testing of cylinders used in transporting gases compressed to pressures over 25 lb. per sq. in. The regulations prescribe certain cylinder-test requirements which must be complied with before the cylinders are acceptable in interstate or intrastate commerce by freight or express. Tests are considered necessary in order to reduce to a minimum the hazard of having in circulation cylinders which are not in proper condition to withstand the service for which they are intended.

The gases which are commonly shipped in these cylinders, and the intervals at which the cylinders must be tested are as follows:

Gas	Intervals at which cylinders should be tested
Acetylene	Once—before putting in service
Anhydrous ammonia	Every 10 years
Carbonic acid gas	Every 5 years
Chlorine	Every 5 years
Ethylene	Every 5 years
Helium	Every 5 years
Hydrogen	Every 5 years
Hydrocarbon gases (other than acetylene)	Every 5 years
Methyl chloride	Every 5 years
Nitrous oxide	Every 5 years
Oxygen	Every 5 years
Sulphur dioxide	Every 5 years
Non-liquefied gases with pressures not over 300 lb.	Once

When cylinders have been tested the date, that is, the month and year in which the test was made, must be stamped into the metal of the cylinder near the top; e.g., a cylinder tested in March, 1925, would bear the marking "3-25."

In order to safeguard the interests of all concerned with cylinders charged with compressed gases, those who use the cylinders should observe the markings showing the dates on which the containers were tested. If at any time cylinders are found which are overdue for test, the owner of those cylinders should be promptly notified that the containers are not tested and marked in compliance with the regulations prescribed by the Interstate Commerce Commission.

In order to insure the greatest safety, cylinders charged with compressed gases should at all times be containers which have been tested in compliance with prescribed regulations. The use of cylinders which have not been so tested, whether in transportation or in service, is unwarranted, and all those dealing with compressed gases should be actively interested in preventing the use of gas cylinders which are not tested as prescribed by the Interstate Commerce Commission.

Boiler-Feedwater Purification

An Inquiry into the Present Status of the Art, and into the Fundamental Considerations Responsible Therefor

By A. S. BEHRMAN,[1] CHICAGO, ILL.

IT IS not to be wondered at if the mechanical engineer, confronted with a problem of feedwater purification, occasionally feels very much at sea as to the best method of solving it. Not that there is any lack of suggested remedies, well recommended—by their sponsors, at least. Far from it. Rather is the engineer in danger of being overwhelmed by the formidable mass of manufacturers' statements concerning various processes or materials, for many of which universal applicability with entirely satisfactory results is claimed in no uncertain terms.

Methods of boiler-feedwater purification, as applied to the major problems of scale, corrosion, and foaming, may be conveniently classified as methods of either exterior or internal treatment.

In the field of exterior treatment the lime-soda ash precipitation process is the oldest. In this process the scale-forming constituents of the feedwater are precipitated, and the precipitate removed by sedimentation and filtration before the water goes to the boiler. "Temporary hardness" in the water—due to calcium and magnesium bicarbonates—is removed bodily. Calcium and magnesium sulphates and chlorides are converted to the corresponding sodium compounds, which are not scale-forming.

A twentieth-century development in exterior treatment is the zeolite or base-exchange method. This consists essentially of filtration of the hard water through a bed of zeolite, as a result of which the calcium and magnesium in the hard water are exchanged for sodium from the zeolite. Zeolite-softened water may be obtained of "zero" hardness, since the conversion of the calcium and magnesium compounds to the corresponding sodium compounds may usually be carried out completely.

Contrasted to exterior methods of treatment is treatment within the boiler itself—usually designated as treatment with "boiler compounds." The number of such compounds is legion. In some cases the goal of scale prevention is sought by introducing some material into the boiler water which will prevent—e.g., by means of "protective colloids"—the deposition of the incrusting impurities in the water as an adherent scale, and keep them in a form that can be removed by blowing down. In others chemicals are added for the purpose of softening the water in the boiler to a greater or less extent, with the formation of insoluble compounds. In still others the object is to coat the boiler metal with a film that will resist the deposition of scale upon it, regardless of the character of the feedwater.

Among more purely physical or physical-chemical devices may be mentioned the open feedwater heater, which is occasionally advocated as a water softener. Mechanical "skimmers" of various kinds have also been advocated with the object of removing periodically or continuously the incrusting impurities from the boiler water as they are thrown down.

Various combinations and modifications of these and other methods have been suggested and employed—the list is too long to attempt to enumerate all of them.

While what has been said in the foregoing applies chiefly to the purification of feedwater for the prevention of scale, it also applies to some extent to the two other major problems of boiler operation—corrosion and wet steam.

What does it all mean? Does it mean that boiler-water purification is such a simple matter that there are a hundred ways of doing it and doing it effectively? If that is true, why spend money for elaborate exterior treatment equipment, when a little compound in the boiler, or a "skimmer" of some sort, or both, will produce the same result? Or is exterior treatment the only really effective and universally applicable remedy, and are all the rest frauds on an unsuspecting public that enjoys being defrauded?

[1] Chief Chemist, International Filter Co.

Presented at a meeting of the St. Louis Local Section of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, St. Louis, Mo., May 22, 1925. Abridged.

It is not the purpose of this paper to attempt to estimate quantitatively the relative values of the various methods of feedwater purification. Its object rather is to inquire into the present status of the art, and into the fundamental considerations that are responsible for that status.

Nearly any kind of treatment will improve a really bad boiler water to some extent. In fact, it is difficult to imagine any treatment devised by a person with even the simplest rudiments of this branch of technology that would not improve a very bad feedwater a little. The treatment might be extremely ineffective and expensive, as many are which have been advocated and used, but it would be strange if some improvement was not observed.

LIME-SODA TREATMENT THE MOST EFFECTIVE METHOD OF FEEDWATER PURIFICATION

The question then naturally arises, what method is generally accepted as the most effective method of feedwater purification?

It is always difficult to generalize accurately; but as far as a general answer can be given, and as far as scale prevention is concerned, the author believes that the majority of authorities will agree on lime-soda exterior treatment as the most generally desirable method. This preference is subject to a number of limitations and exceptions, dependent on local conditions.

There are several reasons for this accord. The process is in itself logical. The carbonate hardness—which in regions of really hard water usually constitutes the bulk of the dissolved incrusting and sludge-forming solids—is removed bodily from the water before it gets to the boiler. The treated water in the boiler is not only non-scale-forming but also carries a minimum of dissolved and suspended solids. If in addition hot-process softening is used, the water is largely, if not entirely, freed from dissolved oxygen, thereby minimizing any tendency to corrosion.

But aside from any inherent advantages, probably the greatest factor in the importance of lime-soda softening is its half-century or more of proven usefulness. Any sense of ownership in the process necessarily disappeared decades ago. The entire process is an open book. The chemical reactions involved have been studied for years in college and factory research laboratories. The chemicals are commercial commodities, bought in the open market. Competition among manufacturers is now concerned only with better means for facilitating the well-understood chemical reactions. There is little opportunity for mystery or commercial "quackery" regarding the process.

It is really this half-century of public study and observation under almost every conceivable condition that has brought lime-soda softening to its present dominating position in boiler-feedwater purification. Lime-soda softening has its limitations, of course, which must be recognized and heeded. Nevertheless it has become generally accepted as the most generally desirable method of treatment in hard-water sections all over the country. Probably the most conspicuous example of this attitude is found in the hard-water regions of the West and Middle West, where the majority of the railroads have accepted lime-soda softening as the most effective method of treatment, and are gradually replacing other methods wherever practical.

THE ZEOLITE PROCESS

The zeolite process of water softening for boiler feed is now beginning to undergo the same sort of trial and scrutiny as did the lime-soda process. The whole subject is just beginning to emerge from the more or less natural domination of the ideas of the first large manufacturer of zeolite equipment. The ten or twelve years in which the process has been used in this country has not been sufficient to give this method its proper and final valuation. That it will have an important place is beyond question. In fact, its value has already been demonstrated in the treatment of water with

low "temporary hardness," such as those of the Atlantic seaboard. In the Middle West and West, hard waters are usually high in "temporary hardness." In the zeolite process this "temporary hardness" in the raw water is replaced by an equivalent quantity of free soda in the softened water; and apart from the effect of this free soda on the tendency of a boiler to foam and prime, the evidence indicates that there is a connection between the free soda and boiler embrittlement. Caution should therefore be exercised in the employment of the zeolite process for the softening of boiler waters of this type.

Internal treatment, scientifically applied, undoubtedly has definite value in certain cases. The use of modified castor oil preparations to reduce foaming induced by soluble salts is a good example. But whatever the value of proprietary "boiler compounds" may be, they will never have the complete confidence and respect of the engineer until the whole subject has been divorced from the absolute dishonesty and fraud admittedly existing in many cases, and in the secrecy and "hokum" too often prevalent even when the question of actual honesty is not involved.

SODA-ASH TREATMENT WITH EXCESS CARBONATE

A tendency to generalize from limited observation is occasionally observable even in the case of well-trained investigators when they make their début in the field of water purification. A pertinent example is the method of scale prevention advocated by R. E. Hall[1] which consists essentially in treating the water in the boiler with soda ash alone, and in providing a certain excess of carbonate for a given quantity of total sulphate in the boiler water, to prevent the formation of calcium sulphate scale. Calcium and magnesium bicarbonates in the raw water are assumed to take care of themselves and to form a non-adhering sludge in the boiler. This sludge may be blown down intermittently, but is preferably removed continuously by circulation and filtration of a portion of the boiler water. It was in conjunction with the manufacturer of such a refiltration device that Hall made his experiments.

Hall's scientific treatment of his subject is much to be commended; and such methods of investigation, when properly and fully carried out, cannot help but contribute materially to the art of water purification. It is unfortunate, however, that, according to the published information available, his generalizations and theory are based on limited and localized experimental evidence.

The "soda ash" or "excess soda ash" treatment of feedwater in the boiler was used a great many years ago on middle-western railroads, and is still being employed on one or two; but the method has gradually been abandoned in favor of lime-soda complete exterior treatment, since the operating economies effected by the latter more than justified the additional cost of installation and maintenance.

The author is not aware that the statement of Hall's theory on the use of a minimum excess of carbonate over sulphate has been previously published as such; and it may be true, as he postulates, that "unless definite attention is given to the maintenance of this ratio it is hopeless to expect to prevent formation of adherent scale." As a matter of fact, however, this ratio has been maintained automatically for a great many years in lime-soda treating plants. It is standard practice to provide one or two grains per gallon (17 to 34 parts per million) excess soda ash in the treated water going to the boiler; and it has also been the practice in dealing with water of the type used by Hall to provide double that quantity or more. Even the smallest-named amount, however, is more than sufficient to provide Hall's typical ratio of 65 parts carbonate to 1000 parts sulphate (at 125 lb. pressure) in practically all usable natural waters, except possibly the particular type employed by Hall. In the railroads using the excess-soda-ash method, the excess carried generally is likewise 1 to 2 grains per gallon.

It is granted that calcium carbonate is not likely to form scale that is hard in the presence of free soda; but if Hall's theory was accurate, it would be difficult to explain the formation of any boiler scale containing any or all calcium carbonate. That such scales do exist is known to every worker who has had many samples of boiler scale come under his observation.[2]

[1] Hall, Fischer, and Smith, *Iron and Steel Engineer*, June, 1924; Hall, MECHANICAL ENGINEERING, Mid-Nov., 1924, p. 810; Hall, *Industrial and Engineering Chemistry*, March, 1925.

[2] E.g., see Paul, Boiler Chemistry and Feed Water Supplies, p. 80 (1923).

Several other statements of Hall also deserve comment. "The pretreatment (with lime and soda)" he says, "is made by the cold process with ample time for precipitation, as separations are the most complete under these conditions owing to the increased solubility of calcium carbonate at higher temperatures." Theoretically, yes. Actually, the residual calcium carbonate is usually about twice as much (or more) in water softened by the cold lime-soda process as by the hot process, due to the incomplete separation of all the colloidal calcium carbonate in the cold, even after very long periods of time.

With waters containing calcium and magnesium bicarbonates (i.e., "temporary hardness"), such as are typical of the hard-water sections of the West and Middle West, the scaling and stoppage of the heated feedwater lines is a serious and expensive nuisance—so much so, in some cases, that its prevention is in itself sufficient justification for exterior treatment. Hall, however, working in actual practice only with acid Pennsylvania waters, dismisses the matter lightly thus:

> It is probably simpler for the feed lines . . . . not to attempt its complete prevention as its removal is exceedingly simple. One method is to bring it in contact with cold water which is saturated with carbon dioxide under pressure . . . . . A quicker and more economical removal of this type of scale can be effected by the use of very dilute hydrochloric acid. Under the rigid supervision of a competent chemist, who would realize the possibility of injury to the feed lines through overtreatment, this method would be very efficacious . . . . . Overtreatment will cause solution of the iron by the HCl.

It is interesting to note that Cross and Irvin,[1] referred to by Hall in his discussion of scale removal with carbon dioxide, stated specifically that they had not found the method applicable with carbonate scales, which are the very ones Hall is proposing to remove; and Jones,[2] to whom Hall also refers, emphasizes the fact that the few results from this little-tried method indicate that it is far from a cure-all. In fact, the idea of using either the carbon dioxide method or particularly Hall's proposed hydrochloric acid method for removal of deposits in pipe lines, will hardly appeal to the experienced engineer or water chemist when applied to the great majority of boiler plants, and particularly the smaller ones, under normal operating conditions.

But the most serious objection to Hall's conclusions is the generalization from a few cases. As far as can be determined from the three papers cited, in his only plant-operating experiments he employed typical Pittsburgh district water—slightly acid (or with very low alkalinity), and with high sulphate hardness. To conclude from this that the same method of soda-ash treatment and refiltration of boiler water would be equally operative with a typical Mid-West carbonate-hardness water, with temporary hardness equivalent to as much as several pounds of calcium carbonate per thousand gallons, is entirely unwarranted. To the author's knowledge, the method described has not been tried out in the Middle West or with waters of this type.

Another recent example is to the point. Because sodium aluminate does unquestionably have value in water treatment as a coagulant in many cases, it does not follow that it will be equally useful in all fields of water purification. The author's attention was called not long ago to a case in California where sodium aluminate was advocated enthusiastically to obviate the effect of a very high sodium chloride content in water to be used in the manufacture of raw-water ice. Such statements naturally lower the informed technical man's opinion of the product, regardless of its real value in certain specific instances.

Instances of all these kinds might be multiplied at great length, but the list would be a long and wearying one. Enough has been said, however, to indicate the liability to excessive enthusiasm on the part of the advocates of a new or untried method of boiler-feedwater purification.

If there is any conclusion or suggestion to be derived from this paper for the engineer who is deciding on a method of boiler-feed purification, it would be the very simple and obvious one that his criterion should be not what is proposed, not what is advocated, but what has actually been shown to be applicable under conditions like his own.

[1] *Power*, vol. 55, no. 11, pp. 422–423 (1922).

[2] *Power*, vol. 60, no. 15, pp. 578–579 (1924).

The Savenius Wing Rotor

A Further Application of the Magnus Effect to Windmills and to Boat Propulsion

By ALEXANDER KLEMIN,[1] NEW YORK, N. Y.

THE Savenius wing rotor is another application of the Magnus effect, whose practical utilization by Flettner has aroused so much interest. In some respects it offers interesting advantages as compared with the Flettner rotor.

The principle of the invention—which is due to Sigurd J. Savenius, a Finnish engineer—is explained in a pamphlet entitled The Wing Rotor in Theory and Practice by means of the two diagrams, Figs. 1 and 2. If two semicircular surfaces are arranged as shown in Fig. 1, the deviation of the air flow is fairly symmetrical and the pressure on upper semicircle is balanced by the vacuum at the back of the lower semicircle. The result is that there is only a moderate torque about the hinge between the surfaces and no pronounced Magnus force at right angles to the air stream. But if the semicircular "wings" are arranged as shown in Fig. 2 with an opening between them, the vacuum at the concave side of wing *a* disappears and is replaced by a region of pressure. There is now a decided torque about the hinge—actually three times greater than with opening closed, the peripheral speed of the rotor instead of being approximately equal to that of the air stream is now approximately

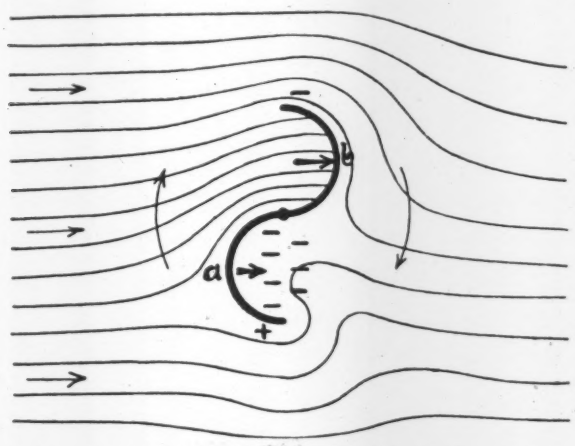

FIG. 1 WING ROTOR WITH PASSAGE CLOSED; NOTE VACUUM AT CONCAVE SIDE OF WING *a*

1.7 times as great, and there is also a very definite displacement of the stream lines above the wing rotor with a powerful Magnus force upward.

The inventor proposes to utilize these properties of the wing rotor in three ways:

1 By using the wing rotor as a windmill with its axis of rotation vertical, or perpendicular to the wind, as shown in Fig. 3

2 By using it as a windmill with its main axis horizontal, a number of wing rotors being mounted radially as vanes on the windmill, and the Magnus force of each wing rotor helping to supply the turning moment of the entire windmill

3 By utilizing the wing rotor in the manner of Flettner's rotor ship, the vessel being driven by the Magnus effect when the wind is on its beam.

Unfortunately the inventor's experiments, while extensive and logical, are more of a qualitative than a quantitative nature. They are interesting, nevertheless, and may be summarized as follows.

THE WING ROTOR AS A WINDMILL

1—Axis of Rotation Vertical or Perpendicular to the Wind. In Fig. 4 are shown some of the forms of the wing rotor utilized in this manner. The openings between the wings were of the order of one-quarter to one-fifth of the spread of the wings, and end plates were employed as in Flettner's design to prevent end flow and loss. The results of comparative tests were as given in Table 1.

[1] Associate Professor of Aeronautics, N. Y. University. Assoc-Mem. A.S.M.E.

TABLE 1 TESTS OF WING ROTORS

	Torque rating
Two-winged rotor with passage	100
Two-winged rotor with passage closed	30
Two-winged rotor with crossed wings	20
Three-winged with passage open	80
Four-winged rotor with inside opening	30

These clearly indicated the superiority of the first form of two-wing rotor acting as a windmill. The inventor then investigated the efficiency of his rotor as compared with a well-designed modern windmill of the type shown in Fig. 5. The windmill had a diameter of 70 cm. (27.6 in.) with 18 vanes, the latter having curved surfaces with a 15-deg. angle at the outer tip and a 44-deg. angle at the base. Two types of wing rotors were tested against the more conventional windmill, No. 1 having a semicylindrical form with the passage one-fifth of the spread, and No. 2 having wings of 165 deg. circumference with the middle opening one-fourth of the spread. The windmill and the two rotors had the same projected area perpendicular to the wind. Numerous tests were made in the open air at varying wind speeds and with varying loads, the

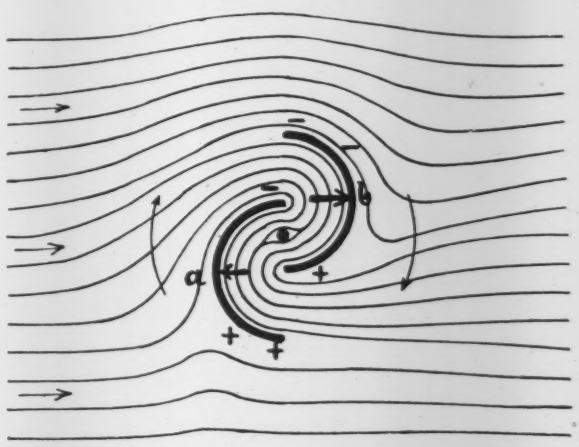

FIG. 2 WING ROTOR WITH PASSAGE OPEN; VACUUM AT CONCAVE SIDE OF WING *a* DISAPPEARS

results of which are summarized in Table 2 (no units are available in the pamphlet).

TABLE 2 COMPARATIVE TESTS OF WINDMILL AND WING ROTORS

Wind motor	Load	R.p.m.	Work	Ratio of power
18-vaned windmill	1.3	390	507	75
Wing rotor No. 1	1.3	520	676	100
Wing rotor No. 2	0.9	705	634	93

It is interesting to note that the wing rotor started under load from any position, even in a feeble wind.

2—Main Axis of Rotation Horizontal or in Line with the Wind. A comparative test was made with a La Cours windmill with four curved vanes. The rotor windmill carried four radial wing rotors each provided with an end disk. The area of the vanes of the La Cours windmill was equal to the projected area of the four rotors. The comparative results were as follows:

	Load	R.p.m.	Work	Power ratio
La Cours windmill	30	110	330	72
Wing-rotor windmill	120	38	456	100

These show that the wing-rotor windmill had a starting torque four times as great as the more conventional windmill, but its speed was much lower.

THE WING ROTOR AS A MEANS OF BOAT PROPULSION

When the wing rotor was tested in comparison with a Flettner rotor of the ordinary type, it gave the same Magnus effect at the

same speed of revolution and airstream velocity. But since the wing rotor is driven by the natural effect of the wind, while the Flettner rotor can be driven by machinery, the latter can be made to produce a greater Magnus effect at will. In slow winds the Savenius wing rotor might have to be provided with an auxiliary motor or else be made very much larger than the Flettner rotor. In strong winds, however, its own turning moment would be sufficient.

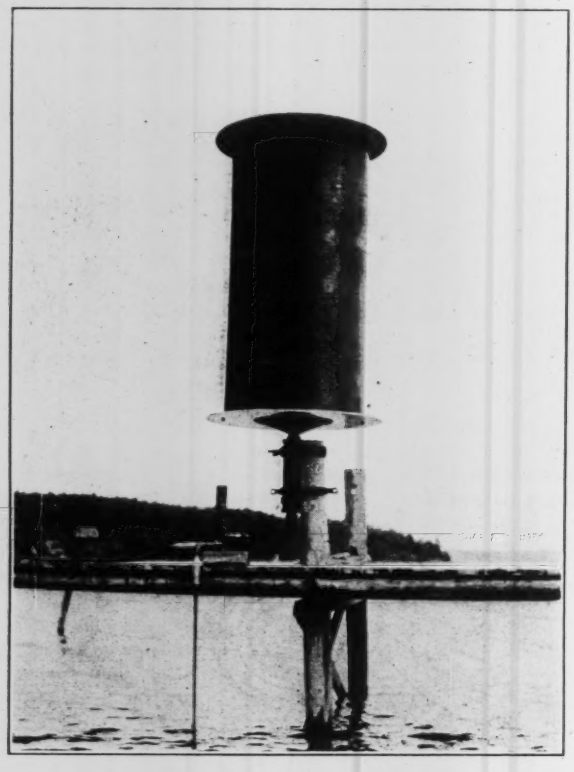

FIG. 3 WING ROTOR AS A WINDMILL OF THE FIRST TYPE WITH ITS AXIS OF ROTATION PERPENDICULAR TO THE WIND

POSSIBILITIES OF UTILIZATION

When used as a windmill of the type first described, the wing rotor appears to offer the following advantages: Somewhat greater power obtainable from the same projected area than with a windmill of more conventional design; ability to extract energy from the wind coming in any direction and thereby dispensing with all vanes, etc.; simplicity of construction; a vertical drive shaft, whereby it

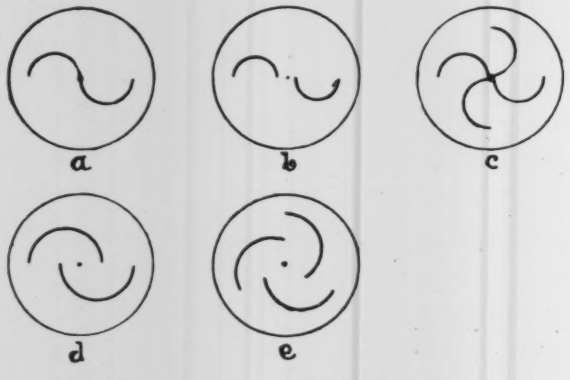

FIG. 4 DIFFERENT FORMS OF WING-ROTOR WINDMILLS OF THE FIRST TYPE TESTED BY THE INVENTOR

is possible to dispense with gearing to a certain extent. Further, by a very simple automatic mechanism it is possible to regulate the size of opening and maintain a constant speed, when needed, with varying wind and load. It should also be possible to vary the characteristics of the windmill at will, and this might be an advantage in receiving power from winds of varying velocity.

The second type of windmill, with wing rotors mounted radically as vanes, does not look so promising. It is more complicated than an ordinary windmill, cannot take advantage of wind from every direction, and appears to be somewhat slow under load.

As a method of propelling ships the idea is interesting, because, unlike the Flettner rotor ship, auxiliary machinery may be dispensed with. Another advantage of Savenius' invention is that the wing rotor offers far more resistance to the wind when stationary than a stationary cylinder, so that it could be used for driving a ship directly with the wind. Also the Savenius wing rotor, by variation of the passages, permits the equivalent of the reefing of a sailing vessel.

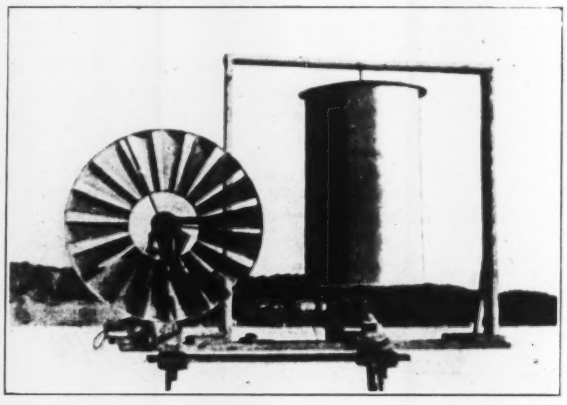

FIG. 5 MODERN 18-VANED WINDMILL AGAINST WHICH THE WING-ROTOR WINDMILL OF THE FIRST TYPE WAS TESTED

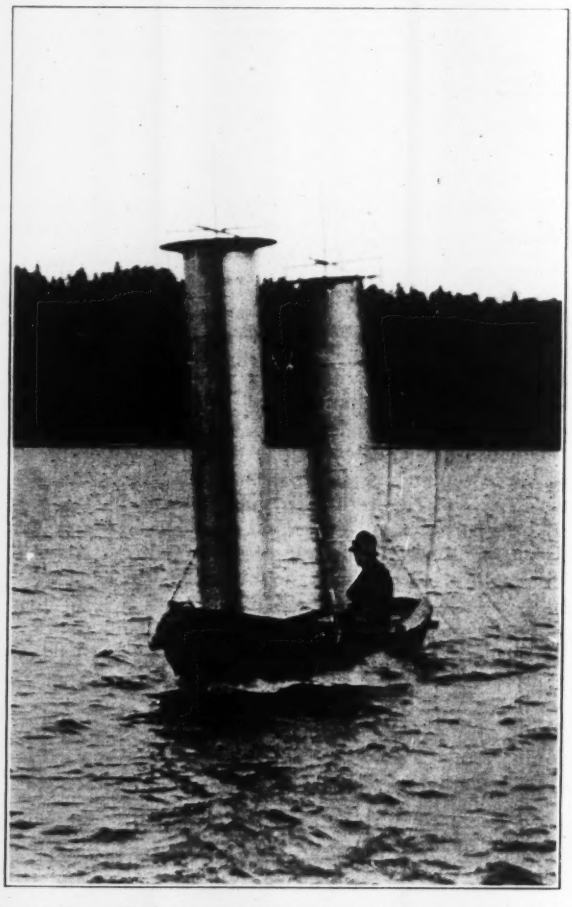

FIG. 6 BOAT EQUIPPED WITH SAVENIUS SAILING AT THE RATE OF 6 KNOTS

Actual tests of the wing rotor in driving a small boat (Fig. 6) and in pumping water have been satisfactory. It would be desirable, therefore, to have more quantitative information in regard to its operation.

Logging Machinery Used on the Pacific Coast

By WM. C. SHAW,[1] PORTLAND, ORE.

IT IS A well-known fact that the introduction of new types of machinery showing radical changes in design, application, or efficiency, does not immediately render obsolete the existing types which have already proven to be suitable for the work for which they have been gradually evolved. There is still a conservative element in humanity, especially engineering manufacturers and users of machinery, who say, "Let George do it," but at the same time, while waiting for the new departure to prove itself, are busy analyzing their own product or equipment to see how much it can be improved in efficiency or adaptation, or still further reduced in price to meet the new competition looming on the engineering horizon. Thus each development in a particular field has a stimulating effect on all apparatus entering that field and even should the type of machine not prove an unqualified success, the effort has been a distinct gain to the trade. Should the new method prove vastly superior to the old, the manufacturer can see the handwriting on the wall and will either "get in step" as quickly as possible or will strike out along another new line of development for himself.

This has been the history of logging engineering from the time of hand logging down to the present day and it will continue so, as this branch of engineering is a product of evolution and the survival of the fittest as much as any other branch of science or nature.

Steam Engines

The steam engine adapted to logging machinery has proven to be a prime mover capable of meeting the exacting conditions in a satisfactory manner.

Fig. 1 shows the wide range in pull and speed from empty drums to full, assuming constant torque and r.p.m. at the crankshaft. The engine is regulated to equalize this to a great extent by throttling or opening out so that it finishes with a heavy overload.

It is paradoxical, but the logging steam engine must be a low-speed, heavy-duty engine capable of withstanding enormous overloads not for a few seconds but for a few minutes; it must also be a quick, snappy, high-speed engine capable of handling light loads or "tripping" back in the minimum of time. Piston speeds in this operation run up to 1500 ft. per min. and over, and unless the engine is well constructed and carefully designed for running balance the vibration is excessive and wear of working parts a constant source of trouble.

This is not so true today as it was in the days of ground yarding, where the engine was often stalled by a log striking obstructions, but still holds good to quite an extent.

Logging-engine designers have evolved engines of rugged construction, with extra-large port areas, late steam cut-off, large wearing surfaces and a splendid running balance, which are today rendering excellent services in conditions where a recognized standard-design engine would be a failure. Totally enclosed, self-lubricating, higher-speed engines have been suggested and the time may be near when these will be tried. Tandem compound cylinders are used in lowering engines, and where there is a long haul will be used more extensively.

Frames

It is somewhat difficult to prognosticate the line of development of any machine, but the process of change for the better still goes steadily forward. Cast iron has been replaced by cast steel in nearly all parts excepting the engine. This has meant quite a reduction in weight, but it is believed that some manufacturers have gone to extremes in this respect. The weight cannot be reduced in anything like the ratio of the relative strength of the two materials, as stiffness and rigidity will be sacrificed, things which are very essential in these machines. Where an open-hearth mild-steel casting is substituted for a cast-iron piece which is subjected to indeterminate stress, a reduction in weight of 30 to 35 per cent is all that is possible. Pieces which are subjected only to tensile stresses can of course be much further reduced.

Many builders are using structural-steel frames entirely and they are proving quite satisfactory where carefully fitted and well riveted. Others use 15-in. 80-lb. steel I-beams, others 18-in. 70-lb. beams with separators or cross-beams of cast steel, semi-steel, or cast iron. These sizes have served very well until recently, when there has been quite an increase in the size and power of logging machinery, and the base for fastening side frames, brackets, etc. being only $5^3/_4$ in. wide at most on the machined top of an I-beam, has about reached the limit for a satisfactory base for the heavier parts coming into use. A probable improvement in this direction will be to rivet and weld heavy bars, say, 1 in. by 8 in., to the tops of these beams and then machine them. This also means vertical bars to stiffen the webs of beams, as they have now reached their limit in lateral stiffness.

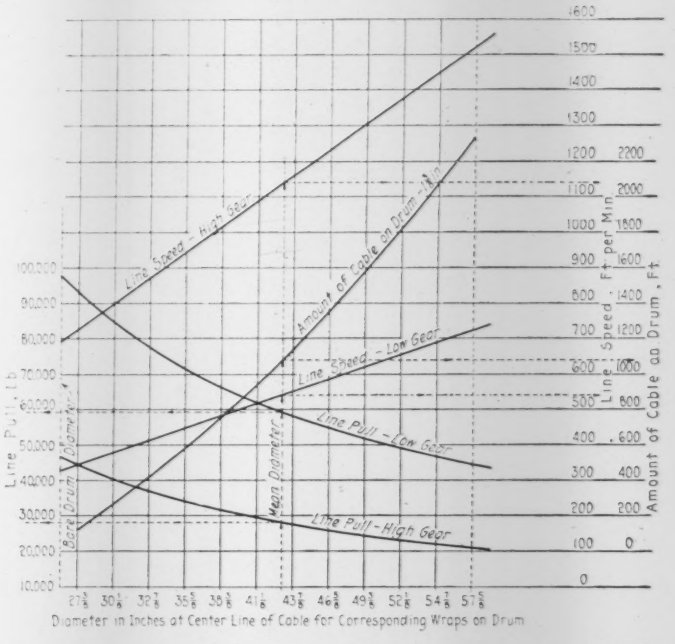

FIG. 1 LINE-PULL AND SPEED CURVES FOR 13-IN. × 13-IN. SIMPLE GEARED TWO-SPEED YARDER
(225 lb. boiler pressure, 350 r.p.m. crankshaft speed.)

For some of the heavy incline engines and skidders coming into the market, double channels or I-beams with covering plate stop and bottom may be used, although this is an expensive form of construction. Where engines are mounted on a railroad car additional stiffness can be given by a proper location of girders under the car deck, and as the side frames are not so high on these special types due to all main leads going to a spar tree, the side stresses are reduced. Theoretically there should be no side stresses, but practically, due to misalignment of bearings from wear, imperfect bearing of gear teeth, or other causes, side stresses always will exist. If the lead up to the spar tree is short there is also considerable side stress on the frames when the main cable is at its extremes of travel on the drum.

The day of babbitted fits in constructing frames and brackets is practically over, and in the future it will be found that all makers will follow the lead of a few pioneers in this field and make their entire construction with interchangeable machine parts, securely tied together with fitted bolts throughout.

Gears and Drums

All gears are now machine-cut with standard involute teeth of apparently massive design. This is an evolution from the ground and high-lead yarder machines as originally mounted on wood sleds. No one yet knows how flexible or how rigid a wood sled

[1] Chief Engineer, Smith & Watson Iron Works. Mem. A.S.M.E.

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is, as a great deal depends on its setting; and gear teeth had to be heavy enough to stand misalignment stresses. The advent of logging from cars which are sidetracked and blocked up has led to a reduction in pitch and to wider faces, thus giving a lighter and smoother-working machine. The author believes that these pitches may be still further reduced if higher-grade steels are used in the castings and (here is the important point) the designer is certain that his sub-frame is rigid and the whole setting as perfect as it can be made. He does not believe, however, that the high-grade helical or herringbone gears now being extensively manufactured for other purposes will be used in logging engine work, as the increased cost could not be balanced in any extra production of logs or elimination of wear in the engine. It is quite probable, however, that a modified stub tooth might be used to advantage in the large heavy-duty machines; and should any manufacturer start an entirely new line of development he will probably adopt this, as the present shop equipment for gear cutting needs only the addition of a new set of cutters for this work.

A probable improvement of the near future will be in friction clutches. The present V-type friction clutch of oak with its numerous variations of fiber inserts, cast-steel blocks with brake-lining inserts, etc., have proven effective, cheap, and easily renewable. There is a demand at present, however, for something better, safer, and more durable, and it is believed that the heavy band-type friction clutch will take its place in new development.

Drums made of cast steel, some made solid, some built up sectionally, have given satisfactory service. Drums built of plate steel, riveted and welded, have been suggested, and where limit of weight is desirable we may see them adopted.

There is now no reason for setting drums on high side frames or brackets as ground yarding machines are practically obsolete. In these older types, the drums had to be elevated above each other so that the path of line from any drum through its fair-leader would not interfere with any other drum. Nearly all leads were horizontal.

With high-lead, skyline, or skidder systems all lines lead up about 60 deg. to the horizontal, and there is no interference in properly arranged drums. The modern tendency, therefore, is to keep all side frames as short in vertical height as possible, even though by so doing the machine may be considerably lengthened. This is a better design and the increased length is no detriment on a sled-mounted machine. On car-mounted machines, however, where a combination of units is made, length has to be given due consideration as cars which are too long are unwieldly and have impaired clearance on the short curves used on logging railroads. Some of these combined units are at present too long, and there is no good reason why they could not be split into two cars, one containing the yarder unit and the other the utility engine and loader. If any more drums are added to the present combinations, this will have to be done.

There is a tendency, at present, to have all operating cylinders for brake frictions controlled by air pressure instead of steam. Condensation troubles are thus eliminated and a more flexible control gained. All heavy machines of the near future will be so equipped.

BOILERS

The present logging-engine boiler is not perfect, but it is astonishing how well it stands up to the excessive demands put upon it in the woods. Were the engines running continuously even under a 50 per cent load, it would not be large enough. However, its reserve capacity is great and it can deliver a large quantity of steam when called for due to the large amount of water contained and the rapid reduction in pressure. Data on boilers are given in Table 1.

Total firebox capacities in the larger sizes of nearly 200 cu. ft. give space for a large fuel bed when using wood fuel and splendid combustion when burning oil. The present working pressures might be increased to 250 lb. per sq. in., but the 225 lb. of today is likely to be maintained for some time.

The principles of combustion are today better understood by the operators, and there is a desire to save fuel and the cost of getting it. Many boilers, pipes, cylinders, etc., are now being lagged with good non-conductors which will effect a 10 per cent per annum

TABLE 1 BOILER DATA

Boiler size	Working pressure, lb. per sq. in.	Water heating surface, sq. ft.	Superheating surface, sq. ft.	Total heating surface, sq. ft.	Grate area, sq. ft.	Ratio water-heating surface to grate area	Ratio max. engine hp. to boiler hp.	Ratio total heating surface to engine hp.
48-in. round	200	307	147	454	9.65	31.8	7.75	1.32
72-in. round	200	937	385	1322	23.7	34.3	5.3	1.82
60-in. single extended	200	735	273	1008	30	24.5	5.9	1.65
72-in. single extended	200	1285	411	1696	41.6	30.5	4.57	2.00
66-in. double extended	225	1222	380	1602	43.5	28	4.75	1.91
72-in. double extended	225	1441	453	1894	47.6	30.2	4.64	1.94

saving in the fuel bill. With proper bricking of the furnace for oil fuel, correct proportioning of the air inlets, and stoppage of all cold-air leaks, a saving of at least 5 per cent can be made over a careless oil installation. Ultimately all of the machines of the large operators will be equipped to burn fuel oil as there are undoubted economies to be obtained should present costs continue. Most large operators have their own railroads, and tank cars can be readily brought to the setting, failing which there is the torpedo tank which may be carried on a skyline or dragged over the ground. This usually carries a day's supply for a yarder or other machine, and can be handled at a time which will not interfere with logging.

Oil-fuel firing is also quite flexible and the demands for steam are quickly responded to by oil burners mechanically controlled from the throttle-operating lever. There are also several successful automatic types on the market. Future types will probably automatically adjust the oil, air, and draft door on a 5-lb. variation in pressure and thus effect additional economies in fuel costs.

Feedwater heaters are being used extensively and will continue to be more popular.

Feed coils in the smoke cone, and exhaust pipes passing through a tank of water, are used extensively, but there will be many improvements along somewhat different lines in the future. Closed heaters using the exhaust from one or both engines will probably be adopted, with automatically regulated feed pumps.

Superheating has been mooted, but the gains from this are problematical. Existing designs of boilers add some superheat to the steam, as at least one-third of the total length of tubes is in the steam space and there is a good transmission of heat from the waste gases through the tube walls.

Locomotive-type boilers are used on some machines, and are much more economical in fuel than the vertical type. In units mounted on cars where there is sufficient space we shall probably see this type come into greater prominence than at present.

DIESEL ENGINES

Mr. Miller in his interesting paper has summed up the case for the Diesel engine.[1] Weights are being reduced, however, on these; higher-speed engines, double-acting, are being turned out at present, and the author believes that when the lumbering industry of the coast has another boom period, we shall see several units such as he mentions being tried out in this section.

Electric logging has been ably taken care of by Mr. Wickes.[2] These installations probably will continue to be installed by the larger lumbering industries whose logging properties are adjacent to their milling operations and who can utilize their waste lumber to produce an excess of electrical energy which can be returned to the woods. The fixed charges on the purchase and installation costs are very much higher than on an equivalent steam-operated setting, and we have still to see a carefully-worked-out balance sheet showing the comparison between a thoroughly modern steam-logging outfit and an electrical one.

The elimination of the fire hazard and the small amount of operating water required are great advantages, but the fire-hazard advantage is being gradually offset by the improvements in oil-fuel burning and better screening methods. The storm hazard in bringing down electric wires has also to be considered, but on the Pacific Coast this is not so great.

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[1] The Overhead-Cableway Method of Logging, by Spencer Miller, MECHANICAL ENGINEERING, July, 1925, p. 527.

[2] Electric Logging, by P. A. Wickes, MECHANICAL ENGINEERING, October, 1925, p. 820.

Steam-Consumption Tests on a Back-Pressure Turbine Built by Brown, Boveri & Co., Ltd., Baden, Switzerland

By A. STODOLA, ZURICH, SWITZERLAND

ONE of the lines of possible development in the direction of improving the performance of steam turbines is in the design of the blading. Our knowledge regarding the relative influence of the shape of the blade, the pitch, the flow of steam, etc. upon the efficiency of the blading is still imperfect. The following report deals with data obtained through the research work of Brown, Boveri & Co., Ltd., in Switzerland.

The machine tested was a 1000-kw. back-pressure turbine fitted with a single-row impulse wheel and twenty-eight reaction stages mounted on a drum.

To enable the efficiency of this blading to be determined separately without the need for further calculations, the impulse wheel was left unbladed and the nozzles were removed so that the steam could flow above and below the rim of the wheel to the reaction drum with the minimum pressure drop. The pressure and temperature of the "working steam" were measured in the wheel casing immediately before entering the reaction portion. The steam leaking past the dummy piston and the gland at the high-pressure end was collected separately and condensed at atmospheric pressure. The leakage at the low-pressure gland was led to the main condenser.

Owing to the smallness of the heat drops they were worked out individually by means of logarithms, using the formulas of Knoblauch. The test results are summarized in a table in the complete paper and are graphically presented in Fig. 1.

The table referred to gives the thermodynamic efficiency ratio referred both to the total quantity of steam supplied to the turbine and to the quantity of working steam actually flowing through the reaction portion. The latter comes into consideration when the efficiency of the blading alone is to be determined. In Fig. 1 the curve of internal efficiency η_i is given; from this quantity the hydraulic efficiency η_s can be obtained by means of the formula[1]

$$\eta_i = (1 + \rho)\eta_s - \zeta_a$$

where $(1 + \rho) = R$, the reheat factor, and ζ_a is the leaving loss, expressed as a fraction of the adiabatic heat drop.

In the present case both ρ and ζ_a are of the order of 1 per cent, so that it may be assumed approximately that $\eta_s = \eta_i$. There are two ways of determining η_i: it can be obtained from the ratio of the "internal" heat drop to the adiabatic heat drop, the former being determined from the actual conditions before and after that portion of the turbine under consideration; alternatively, η_i can be obtained from the ratio of the measured internal (or "indicated") output to the calculated output of the perfect turbine. The internal output is the sum of the output at the coupling and the power absorbed in bearing friction, in driving the governor and oil pump, and in the disk friction of the unbladed impulse wheel. The last-named quantity must be included in this instance, as it occurs before the point where the steam enters the reaction portion, and so may be considered as an addition to the bearing friction. It is calculated from the empirical formula—

$$N = 10^{-6} \times 0.256 \times D^{4.2} \times \left(\frac{n}{1000}\right)^{2.8} \times \gamma$$

where N = output in kilowatts; D = diameter in inches; n = speed in r.p.m., and γ = specific weight of the steam in lb. per cu. ft., which formula has been established from observations made by Messrs. Brown, Boveri & Co., and gives more accurate results than the corresponding formula quoted in the author's book.

A comparison of the values of η_i determined in these two ways shows that the one obtained by consideration of the mechanical losses is throughout somewhat higher than that obtained from the steam conditions (in Fig. 1 indicated by "from O" and "from T," respectively); nevertheless, considering the smallness of the output concerned, the two values are sufficiently close in agreement for practical purposes. The accuracy of the value determined from the mechanical losses is certainly greater than that based on temperature measurements, and this fact was taken into account in drawing the graphs.

An important factor to consider in judging the results obtained is the radial clearance of the blade tips, and special attention was given to its measurement. Regarding the radial clearance of large turbines, Brown, Boveri & Co. advise that in the older types of single-cylinder turbines, in which the ratio of the distance between bearings to the blading diameter is large, a clearance equal to one-thousandth of this diameter has proved sufficient allowance for all the variations which may occur under the conditions of operation. In the case of the most recent two- and three-cylinder turbines the distance between the bearings is comparatively smaller, so that the clearance at the blade tips can be reduced to 0.0008 of the diameter. When the low-pressure rotor is of the disk type the clearance is great, as the guide blades project freely into the spaces between the disks, the blading and shrouding being so arranged that gaps are "bridged over." In this way practically the whole of the steam jumps across the gap, and the loss is said to be kept down to 3 per cent of the theoretical output. These particulars enable the problem maximum efficiency of such a multi-cylinder machine to be estimated (see summary).

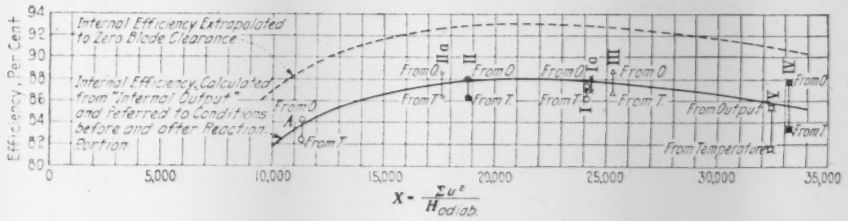


FIG. 1 THERMODYNAMIC EFFICIENCIES AS A FUNCTION OF THE PARSONS COEFFICIENT

(The Parsons coefficient X = sum of squares of peripheral velocities in meters per second, divided by the adiabatic heat drop in kilogram-calories per kilogram.)

From other data it would appear that a 1 per cent increase in blade-tip clearance causes the efficiency to be reduced 3.1 per cent. If this figure be applied to the machine at present in question, and it be assumed that $\eta_s = \eta_i$, the efficiency of the blading with the blade-tip clearance reduced to zero will be increased by $3.1 \times 1.62 = 5.0$ per cent, so that $\eta_s = 93.0$ per cent, as shown by the dotted curve of Fig. 1.

A comparison of the new efficiency curve with the older curve appearing as Figs. 710 and 710a in the author's Dampf- und Gasturbinen, 6th edition, shows the extent of the progress which has resulted from the introduction of the new blading.

Upon the conclusion of the tests described above, the impulse wheel was bladed and the complete turbine tested. As the purpose of these tests was to bring into prominence the efficiency of the essential working element of the turbine, i.e., the blading, the efficiencies have been calculated on a basis of the steam conditions ahead of the nozzles, and not ahead of the main stop valve. In this way the results are entirely independent of the dimensions which happen to be selected in designing the steam passages of the main stop valve. The steam leaking past the dummy piston was again condensed separately, thus enabling the internal efficiency to be determined with greater accuracy.

The results of these tests are given in the form of a table in the original article, and from this it would appear that the internal efficiency of the reaction portion agrees well with the values obtained in the first series of tests. The efficiency at the rim of the impulse wheel is 77 to 79 per cent, which is not as high as one could expect from good single-row wheels. The drop of the internal efficiency of the impulse wheel (calculated from pressure and

[1] Stodola, Dampf- und Gasturbinen, 6th edition, p. 207.

temperature) to 71.5 to 74.1 per cent tends to indicate that with steam admitted to only one-third of the circumference, the steam at 107 to 114 lb. per sq. in. carried round by the blades interferes to an abnormal extent with the flow of steam from the nozzles.

Summary

The results given at the top of the following column were obtained in Baden, Switzerland, on a Brown, Boveri back-pressure turbine of about 1000 kw. and consisting of one impulse wheel and 28 reaction stages.

The last figures show that a considerable advance has been achieved in the form given to the reaction blading, which opens up great possibilities for the construction of extremely economical turbines. It is all the more noteworthy that these figures were obtained with a turbine of relatively small output, the blade-tip clearance being sufficiently large for the turbine to be guaranteed as safe and reliable in service.

Efficiency ratio of the whole turbine referred to the steam conditions ahead of the nozzles and in the exhaust chamber, and to the *output at the coupling* for the *working steam* only.....82 per cent
Efficiency ratio as above but for the *total quantity of steam*.....80 per cent
Hydraulic efficiency of the reaction portion alone, referred to the steam conditions before the reaction portion and in the exhaust chamber, and to the *internal* (or "indicated") *output* for the working steam only (0.73 mm. = 0.0287 in. radial blade clearance).....88 per cent
Hydraulic efficiency as above, but extrapolated to zero blade clearance.....93 per cent

The extent of the improvement can be estimated numerically by thermodynamic calculations, and an efficiency of 86–87 per cent at the coupling appears to be possible. (*Zeitschrift des Vereines deutscher Ingenieure*, vol. 69, no. 37, Sept. 12, 1925, pp. 1177–1181.)

American Society for Steel Treating

Abstracts of Papers Presented at Cleveland Convention of Special Interest to Mechanical Engineers

THE seventh annual convention of the American Society for Steel Treating, held in Cleveland during the third week of September, covered a range of subjects that extended far beyond what was apparently the original purview of the Society, namely, steel treating; for not only were important and interesting papers on steel melting, general metallurgy, physics, etc., presented and discussed, but in addition there were two engineering sessions, one in coöperation with The American Society of Mechanical Engineers and the other with the Society of Automotive Engineers. Some of the most important papers from the point of view of the mechanical engineer are briefly abstracted below.

Oil-Burning Equipment for Industrial Furnaces

M. H. Mawhinney, Mem. A.S.M.E., Engineering Dept., General Furnace Co., Philadelphia, Pa., presented a strictly practical paper on this subject, in which the installation and operation of oil-burning equipment are reviewed, starting with the storage tank and including the auxiliary equipment. It covers in a general way the chief points of trouble encountered in supplying fuel to the furnace.

The author points out that the arrangement of the bricks inside the furnace must be correct for the satisfactory burning of oil. The principal considerations are the burner, holes, flues and ports, combustion volume, and baffles.

The troubles encountered in lighting a cold furnace are discussed and the correct procedure to follow in this operation is given; and the location and remedy of troubles, other than purely mechanical failures, are discussed at some length.

It has been found by tests conducted by the United States Navy that for best atomization in fuel-oil burners the viscosity of oil should be about 8 on the Engler scale, or about 300 Saybolt seconds. The temperature to which oil must be heated to obtain this viscosity depends on the gravity of the oil, but for any oil lower than 27 Baumé gravity, this temperature must be greater than room temperature. For an average oil of about 18 Baumé gravity, the temperature required is about 130 deg. fahr. It is therefore advisable in most cases to install an oil heater between the pump and the burners. This heater can be one of many standard types, and gives little difficulty.

Carburization and Case-Hardening

These subjects received considerable attention at the meeting. W. E. Day, Jr., Metallurgist with the International Motor Co., New Brunswick, N. J., presented a paper dealing with the carburizing process when using solid compounds, and considering the points which have the greatest bearing on the commercial application of the process. He discussed the production of carburizing gases from the cement and their reaction on the steel. The quantity of hydrogen evolved in carburizing, he said, is greater than that which could have been produced by the decomposition of the water which primarily existed as moisture. It is therefore highly probable that it is at least partly due to the decomposition of hydrocarbons. The action of hydrogen cannot be disregarded for it is undoubtedly because of its presence that the nature of the action of solid compounds and hydrocarbons varies so greatly from the action of pure carbon monoxide. Furthermore, hydrogen is under certain conditions a strong decarburizer.

Since it appears that the nature of the gases evolved from the compound vary greatly with even a slight change in the mixture and that the character of the charcoal is also of vital importance, it follows that careful checking of the compound is highly essential. This does not mean that it is necessary to make long and tedious chemical analyses, for in fact, unless such analyses are much more than what we may term commercially complete, nothing will be gained. An analysis of the evolved gases at working temperature would, perhaps, be useful, but due to the complexity of the reactions, such analyses may be misleading. However, since that with which we are entirely concerned is the ability of the carburizer to furnish carbon to the steel, it appears that the best test would be to ascertain under standard conditions the action of the carburizer on the steel itself. This can best be done as follows: A bar of steel of the suitable analysis is turned down and cut off to form a cylinder some three inches in diameter and six inches long; it is then packed in a small standard box with the carburizer to be tested and heated carefully to a predetermined temperature for a certain time; the box is then removed from the furnace and allowed to cool. After the test piece has been thoroughly cleansed of any adhering particles of compound, five cuts each 0.006 in. deep are taken on the lathe and the carbon in each consecutive layer determined by combustion in the regular manner. This procedure sets up a standard for future checking.

The author discussed next the action of the gases on the steel on the mechanism through which the carbon is conveyed from the surface inward. He came to the conclusion that although the rate of diffusion is dependent upon both the value of the constant of diffusivity and the difference in concentration, the greatest effect is due to the great change of the constant of diffusivity K with changes in the temperature, it being impossible for the difference in concentration to vary widely because of the limited solubility of the carbon in iron. The true relation of time and temperature to depth and nature of the case cannot, however, be expressed by any simple relation, and it is only by the close observations of several conditions that the desired type of case may be obtained.

The Interpretation of Notched-Bar Impact-Test Results

In a paper on this subject, Paul Heymans, Mem. A.S.M.E., Assistant Professor of Physics, Massachusetts Institute of Technology, reported on some recent investigations he has made by means of the photoelastic method of stress distribution around notches used in impact tests. The shape and dimensions of the notches were analyzed under uniform static longitudinal pull and below the elastic limit.

Stress distribution as a whole, and also the maximum stress and the ratio of the maximum stress to the mean stress, he said, vary widely with the different types of notches. Similar variations will exist when specimens representing those notches are put under bending, as they are in the Charpy impact test. It is true that the relative values of the stresses, as obtained below the elastic limit, will be somewhat modified when the elastic limit is exceeded, but the stress distribution will be qualitatively maintained and failure will start at the point of maximum stress and depend upon the value of this maximum stress below the elastic limit and not upon the value of the mean stress or any such expressions.

The nature of the local failure at the point of maximum stress and its propagation, the author states, will depend upon the general distribution and also upon the elastic properties of the material. If the material is ductile, a localized excessive stress may only result in local yielding, although if the material is not sufficiently ductile it may give rise to a crack.

STRESS-STRAIN CURVES AND PHYSICAL PROPERTIES PERTAINING TO HARDNESS

In a paper bearing the above title, H. P. Hollnagel, Physics Department, Thomson Research Laboratory, General Electric Co., West Lynn, Mass., discussed the various physical properties of materials with reference to their atomic structure and changes occurring therein. Fig. 1 is supposed to give a stress-strain curve showing different kinds of deformation, of which the author claims there are three: elastic, hysteretic, and permanent.

The first of these causes a displacement of material in conformity with Hooke's law and is viewed as being the result of an attempted alignment of the different aggregates along the line of stress. This results in an interference at intercrystalline planes causing interpenetration of atoms during the stress, with elastic return consequent on the removal of that stress.

As the crystalline masses become more lined up in their respective orientation, slip takes place to a greater degree, with consequent gradual decrease in the effect of elastic interfering forces. Directly above the elastic limit there exists a transition region in which material after being stressed, if permitted to rest, tends to relieve its strain and return to its original state of perfect elasticity. As a matter of fact, however, it never does completely return to that state.

The origin of the hysteretic and permanent deformations is derived in approximately the same manner.

The three points characterizing a stress-strain curve are: (1) the proportionality or elastic limit, P; (2) the yield point, Y; and (3) the ultimate limit, U. In the light of what has been said, they have the following significance: P represents a point in the strain history of a specimen of metal, at which slip on atomic planes outweighs the transverse forces of restitution previously referred to. In a perfectly elastic material no such point exists, its restoring forces are always 100 per cent effective; whereas for an imperfectly elastic substance, at this point a sufficient volume of material changes its configuration, in the aggregate, causing the average restoring moments of the forces of restitution to be insufficient for immediate recovery to the original state.

The point Y indicates that slip on atomic planes in mass has so far outweighed the forces of restitution that the configuration of separate aggregates forming a certain volume of the test piece has been definitely distorted or destroyed. The forces of restitution of the individual aggregates have therefore become ineffective. U records the point at which restitution is zero and the atomic slip in mass has proceeded to such an extent that the so-called "keying" action between slip planes has caused their destruction. Such a volume of the material has been destroyed that shearing is effected at the crystal surfaces, with the result that rupture occurs.

The author then defines various properties of materials, such as brittleness, ductility, resilience, plasticity, toughness, etc., in terms of P, Y and U. When P, Y and U are of such values as to be practically indistinguishable, a substance is said to be brittle. Atomically speaking, this requires that the atoms or aggregates be so firmly packed and interlocked that the condition of internal strain is imminent under the slightest displacement from normal physical conditions. A locally applied stress in such cases causes large internal strains. On the other hand, when Y is removed from P

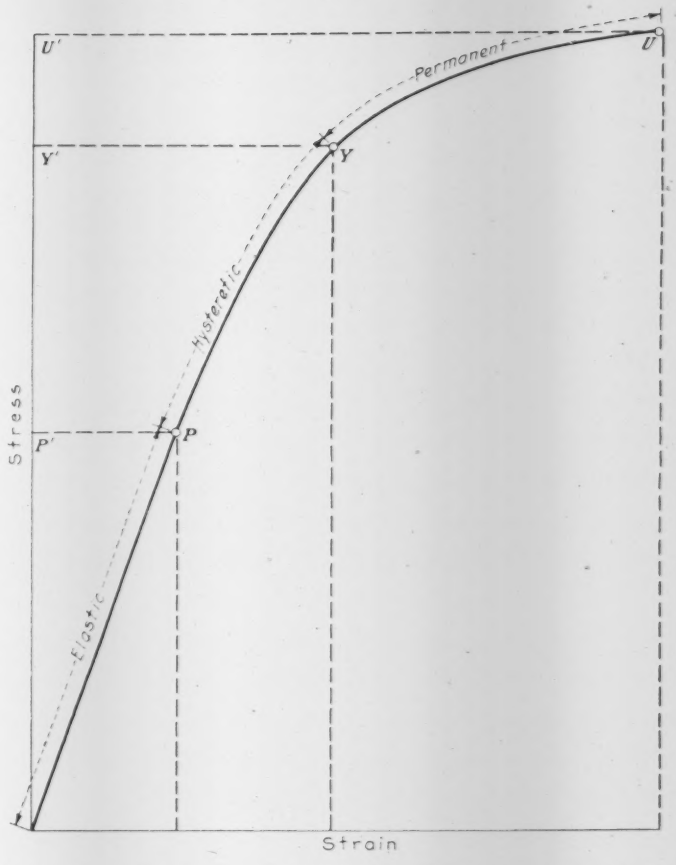


FIG. 1 STRESS-STRAIN CURVE SHOWING THE DIFFERENT KINDS OF DEFORMATION

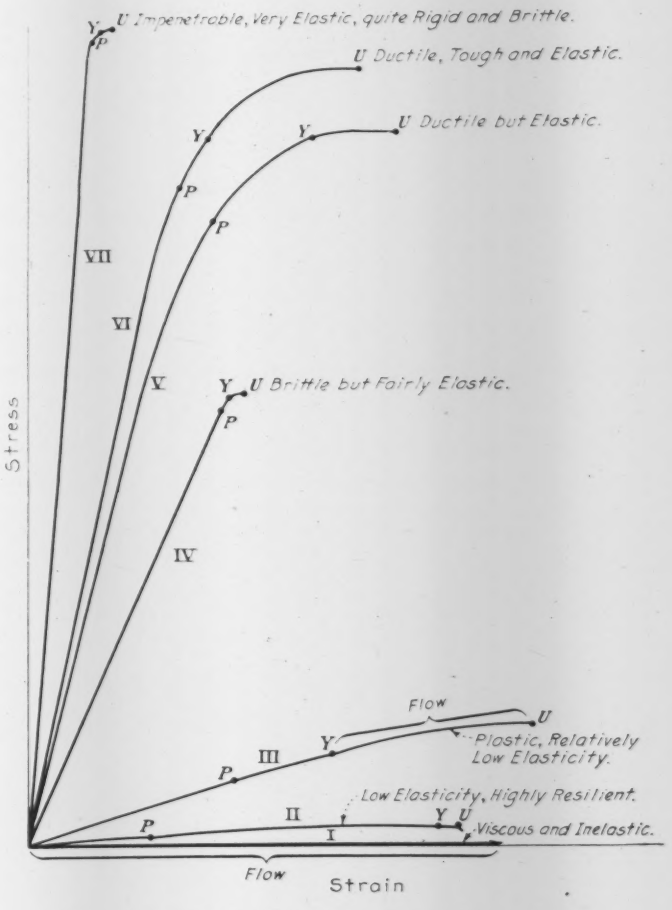


FIG. 2 DIFFERENT TYPES OF STRESS-STRAIN CURVES

and U is markedly distant from Y, elastic after-effects, though they cause a gradual readjustment of the test piece, have an action so slight compared to the change which permits of permanent distortion of the specimen that it is relatively insignificant, with the result that the material is ductile.

Resilience. A body which when stressed has its dimensions considerably altered but on release of the stress resumes its original dimensions or very approximately so, is said to be resilient. This means that in the stress-strain diagram, Fig. 2, the distance between P and Y may be relatively large and that between Y and U small.

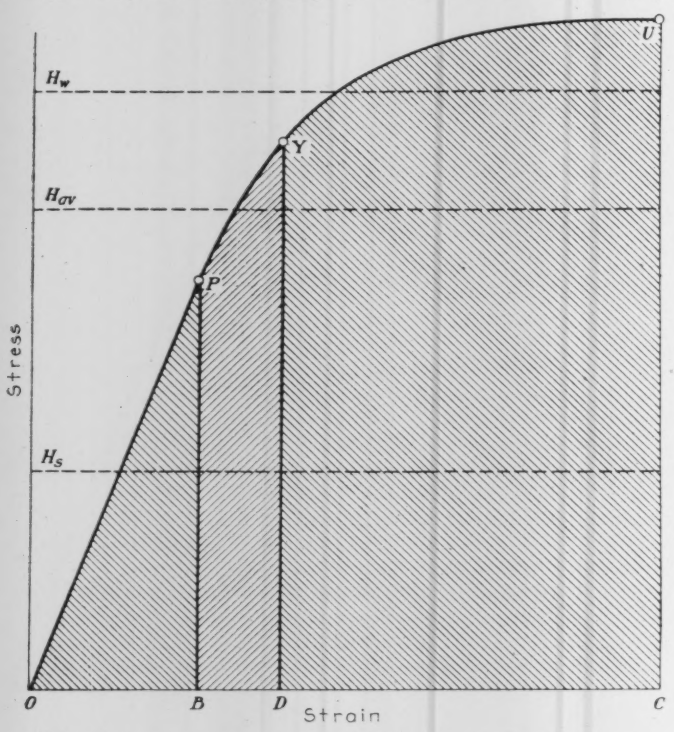


FIG. 3 STRESS-STRAIN CURVE AS USED TO DEFINE HARDNESS

It also means that such material as is present in the lattice form may slip along the direction of atomic planes, but is insufficient in volume compared to that which acts to interfere with the aligning action of the stressing forces. When, however, the latter forces have reached sufficiently great values, the material not only slips suddenly in the mass but ruptures almost simultaneously at intercrystalline surfaces.

Rigidity and Hardness. If a stress, whatever its magnitude, produces zero strain in a body, the modulus of that material would be infinitely great; it would be perfectly rigid and would possess neither yield point nor ultimate limits for they would be infinite in value. In the case when a minute force produces a very large strain the modulus would be zero, the elastic limit zero, there would be no rigidity, and again, but for a definite reason, no yield point or ultimate limit.

All engineering substances lie between these two extremes. The most impenetrable and the softest materials have infinite and zero moduli, respectively, but the modulus alone is insufficient to characterize an engineering material from a technical standpoint and in fact, no one property can be used to characterize a material properly. It must be defined in terms of effects and depends upon the chemical (atomic and crystalline) and physical (microscopic and macroscopic) constitution of the material.

The average property which includes the effects of all other characteristics is hardness. It is an average resistance of the material to deformation of any kind. This hardness may be divided into two different kinds, surface hardness and hardness of cold work, but in either case the hardness represents a physical resistance due to an average of the resultant forces of restitution of that volume of material which is immediately affected by the working stress. It may best be defined in terms of the stress-strain curve as this gives us a record of the necessary physical facts for a particular piece of material. (See Fig. 3.)

The area under this curve is indicative of the energy required to stress a certain volume of the substance to the ultimate limit. It is characterized by three parts, OBP, $BPYD$ and $DYUC$. The first is the deformation energy necessary to stress the material at the surface to the elastic limit; the second, the energy which characterizes the work of deformation beyond the proportionality limit to the yield point and which may be termed the energy of hysteresis; the third, the energy which characterizes cold work, i.e., between the yield point and ultimate limit. If, therefore, these be summed, there results the energy used in overcoming three effects. This acts on a certain volume of material, developing a strain. If distributed uniformly over the entire strain, a quantity results proportional to all its engineering properties. It therefore appears that solid substances have a particular property characterized by the work done on them per unit volume per unit strain for the applied stress. This property has different values, depending entirely upon what portion of the stress-strain curve is examined. At P, for example, its value is $\frac{(OP)(OB)}{2(OB)} = \frac{OP}{2}$. At Y it will have the value given by $\frac{\text{area } OYD}{OD}$. In this manner a value is obtained which is entirely characteristic of the entire stress history of the material.

The author claims that solid substances have a property characterized by the work done on them per unit volume per unit strain for the applied stress, and that this property has different values, depending entirely upon what portion of the stress-strain curve is examined.

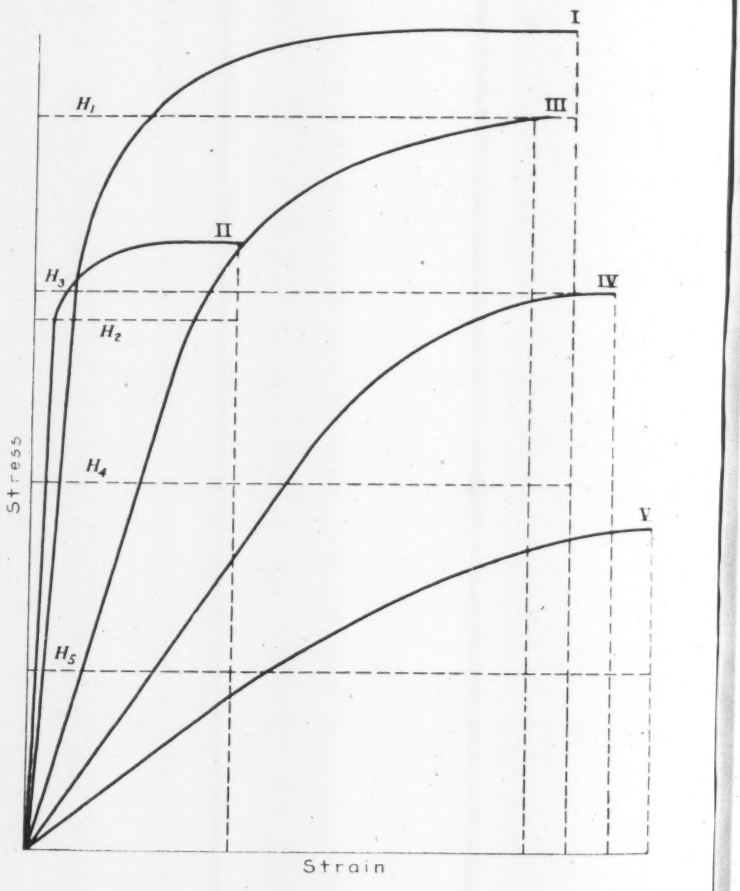


FIG. 4 VARIOUS KINDS OF STRESS-STRAIN CURVES AND THEIR AVERAGE HARDNESS CHARACTERISTICS

The author's "average hardness" (Fig. 4) is therefore a definite physical quantity having the dimensions of pressure $ML^{-1}T^{-2}$, being the work necessary per unit strain, per unit volume, to overcome the average of the forces of resistance of all the atoms participating. This in the last analysis characterizes the different hardness numbers herein referred to. It is termed the "average hardness" of the material not only because it is an average of the atomic

forces of restitution with which we are concerned, but also because the work of deformation is cumulative in producing any resultant strain. This, when divided by that strain, must give an average value. It is equivalent to constructing a rectangle of such a height as to give the same area as that under the curve having the strain as its base.

It provides a possible means for comparing various types of hardness numbers. These are determined by devices which do work while elastically or permanently deforming material in a non-uniform manner; and as there is a stress-strain diagram characteristic of every solid of given dimensions, a ready comparison of their numbers resolves itself into a determination of the volume of stressed material and the magnitude of the strain.

From this the author proceeds to a discussion of hardness as measured by various devices and to the correlation of hardness numbers; the aim of this part being the theoretical unification of all hardness measurements.

In connection with this question of measuring hardness another question came up at the meeting, namely, the relation between hardness and machinability. Practically all the speakers were apparently inclined to the belief that there is no such direct relation. It was pointed out that when two pieces shaped like tool bits were made, one of aluminum alloy of Brinell hardness of about 150 and another of Armco iron of hardness 85, it was the softer bit that made an imprint on the harder one and not vice versa. It was also pointed out that one of the manufacturers had developed a steel which, notwithstanding that it had a Brinell hardness of 650, was machinable.

Other speakers raised the question as to just what machinability is, claiming that this in itself has not yet been acceptably defined. It was stated that there are several kinds of machinability, such as, for example, the removal of metal from a piece in a machine tool or the production of a smooth finish. Taking aluminum and its alloys as an example, there is no trouble in taking a heavy cut with a proper tool, but when it comes to the question of producing a smooth surface, troubles begin. In other metals the reverse condition prevails. The general impression which was left by the discussion, however, was that thus far no direct functional relation has been established between such hardness as is, for example, measured by the Brinell tester and machinability in the sense of ability to remove metal and produce a decent finish without excessive wear on the tool.

Effect of Cold Working on Hollow Cylinders—Elastic Strength Increased by Enlargement of Bore

Dr. F. C. Langenberg, of the Watertown Arsenal, Watertown, Mass., presented a paper on this subject in which he discussed the distribution of stresses in a cylindrical structure subjected to internal pressures such as large guns, and described the equipment used in performing the tests. He pointed out that with the steel in a certain condition a given percentage of enlargement of the bore results in unequal elastic strength which apparently is not affected by the mass of metal in the cylinder wall as long as the ratio between the inside and outside diameters is a constant; i.e., a cylinder having a 3-in. bore and a 9-in. outside diameter has the same elastic strength after a given enlargement as a cylinder having a 6-in. bore and an 18-in. outside diameter, assuming that both are made from the same grade of steel.

In speaking on the effect of various annealing operations on the elastic strength after different amounts of cold working had been applied, the author cited a cylinder which behaved elastically up to a pressure of 50,000 lb. per sq. in. After an enlargement of 3 per cent had been made, the cylinder behaved elastically up to a pressure of 80,000 lb. per sq. in., showing an increase of 30,000 lb. per sq. in. The cylinder then was annealed at 482 deg. fahr. This annealing increased the elastic strength of about 7000 lb. per sq. in. The cylinder then was given an additional half per cent enlargement and showed a still further increase in the elastic strength. The curve which the author used on the screen in this connection showed that the increment in the elastic strength produced by the additional 0.50 per cent enlargement after the annealing operation was much greater than the increase would have been if the cylinder had been enlarged from 3 per cent to 3.50 per cent without an intervening anneal.

Mechanics of Failure of Metals

H. F. Moore, Mem. A.S.M.E., in charge of the Investigation of Fatigue of Metals, University of Illinois, Urbana, Ill., in a paper on this subject discussed what happens when metal fails by fatigue under repeated stress. He pictured two actions going on in a metal under repeated stress, one a strengthening action and the other a destructive action, the former an increase in resistance to slip between crystalline formations and the other a gradual "filing" action tending to start and spread minute cracks in the metal. If the internal force in the metal (stress) is low, a slipping within a few "unfortunate" crystalline grains occurs, but the increased resistance to further slip in these grains throws the stress to other grains and a readjustment occurs without much destructive "filing" action, and as a result of the whole action the metal is stronger than it was at first. This is in agreement with test results in which after 100,000,000 cycles of low stress had been applied to specimens of mild steel the strength to resist reversed stress was found to be distinctly greater than was the strength in the case of unstressed mild steel.

If the repeated stress is very high the slipping action affects many more crystalline grains, and instead of dying out after a few thousand cycles of stress the destructive "filing" action keeps up until there is formed a crack visible under the microscope, or sometimes visible to the unaided eye.

J. F. Keller, in Charge of University Extension Work, Purdue University, Lafayette, Ind., in a paper entitled, Why Metal Warps and Cracks, discussed the unequal heating and cooling of metal and showed how deformation due to these factors was one of the important causes for the warping and cracking of metals. The original paper contained a graph showing the loss of strength due to the heating and discussed in detail the various causes where such unequal cooling or heating made trouble.

Other Papers Presented

B. D. Saklatwalla and H. T. Chandler, both of the Vanadium Corporation of America, presented an interesting paper on the Application of the Mathematics of Probability to Experimental Data as a Basis for Appropriate Choice of Ferrous Materials. To do this the authors use a general curve expressing the so-called law of error, in which the extent of the deviations from a supposedly uniform course are shown as well as the frequency with which they occur.

Probability has been defined as the measure of the strength of our expectancy of a future event. Two classes of steel may have similar specifications as derived from laboratory tests, but their probability curves may show different results as to their chances of failure and success in the actual service under all existing commercial conditions. Under laboratory conditions it is possible to obtain with carbon steel physical properties fully equal to those of alloy steels. Under actual conditions, however, the chances of obtaining the service results with alloy steels are so much greater than with carbon steels that preference is given to the former. The same reasoning is applied to drawing a probability curve in which Brinell hardnesses are plotted against the frequency number of castings in foundry practice. It would appear that this graphic method of representation tells a different story from what mere averages of the numbers would give, and the probability curve appears to give indications of greater value than do average numbers.

In the session contributed by the Machine Shop Practice Division of The American Society of Mechanical Engineers two papers were presented, one by H. K. Spencer, Blanchard Machine Co., Cambridge, Mass., on Modern Surface Grinding, and the other by S. A. Keller, Keller Mechanical Engineering Co., Brooklyn, N. Y., on Dies. Both were of a general and practical character.

In a paper on Graphitization at Constant Temperature, H. A. Schwartz, Mem. A.S.M.E., Manager of Research, National Malleable and Steel Castings Co., Cleveland, Ohio, described a means for the laboratory determination of graphitizing rate as a physical constant of any given hard iron. He showed that this rate is determined by the rate at which carbon can migrate in iron. The progress of graphite formation with time is shown to be an expression of the changing migratory distances and concentration gradients produced by the reaction.

Code on Instruments and Apparatus

Preliminary Draft of Chapter 21, Dealing with Condenser-Leakage Tests

THE Main Committee on Power Test Codes takes pleasure in presenting to the members of the Society for criticism and comment Chapter 21 of the Code on Instruments and Apparatus, dealing with Condenser-Leakage Tests. The Individual Committee which developed this draft consists of Messrs. C. F. Hirshfeld, Chairman, C. M. Allen, E. G. Bailey, L. J. Briggs, W. A. Carter, J. D. Davis, R. E. Dillon, F. M. Farmer, J. J. Flather, J. B. Grumbein, O. Monnett, S. A. Moss, R. J. S. Pigott, E. B. Ricketts, S. W. Sparrow.

In 1918 the Committee on Power Test Codes was organized by the Council of the A.S.M.E. to revise and enlarge the Power Test Codes of the Society published in 1915. This Committee consists of a Main Committee of twenty-five members under the chairmanship of Fred R. Low and nineteen individual committees of specialists who are drafting test codes for the various prime movers and the other auxiliary apparatus which constitute power-plant equipment.

The Individual Committee, the Main Committee, and the Society will welcome suggestions for corrections or additions to this draft from those who are especially interested in this subject. These comments should be addressed to the Chairman of the Committee in care of The American Society of Mechanical Engineers, 29 West 39th Street, New York, N. Y.

TABLE OF CONTENTS

PART I—CONDENSER LEAKAGE IN GENERAL

	Par.
General	1
Qualitative Silver Nitrate Method	5a
Quantitative Silver Nitrate Method	5b
Short Direct-Weight Method	5c
Standard Direct-Weight Method	5d
Electrolytic Conductance Method	5e

PART II—PROCEDURE FOR TESTS

Collection of Samples	6
Qualitative Silver Nitrate Method	20
Quantitative Silver Nitrate Method	26
Short Direct-Weight Method	39
Standard Direct-Weight Method	54
Electrolytic-Conductance Method	62

PART III—SPECIFIC INSTRUCTIONS

Fresh Circulating Water	89
Salt Circulating Water	93

PART I—CONDENSER LEAKAGE IN GENERAL

1 Whenever a steam-rate test of an engine or turbine is to be conducted by weighing or measuring the condensate discharged by a surface condenser, the amount of leakage of circulating water into the steam space of the condenser shall be determined by one of the methods given below, and, if it exceeds the value specified in the Test Codes for Steam Turbines or Reciprocating Steam Engines, shall be reduced to such value in order to establish satisfactory test conditions for the engine or turbine. (The methods given do not apply to jet condensers.)

2 The steam rate of the engine or turbine, of course, shall be corrected for the average leakage shown by the leakage tests.

3 It is to be noted that the amount of water or steam used in sealing glands, valves, and joints of the engine or turbine, hotwell pump, and other auxiliaries is not condenser leakage. Any external water used for such purposes must be accounted for independently of condenser leakage, during both the steam-rate test and condenser-leakage test. The arrangements in all such matters must, of course, be the same during the condenser leakage test as during the steam rate test. (See Test Code for Steam Turbines, Par. 29.) As there specified, it is highly desirable to use condensate taken from the line between the hotwell pump and the weighing tank for all sealing purposes. If this is not done and if water from some external source is mixed with the condensate, there is the additional complication of keeping track of such water, as well as an error, due to the different condition of such water, when the electrolytic or silver nitrate method is used for determining the amount of condenser leakage.

4 There are five permissible methods for determining condenser leakage which are described briefly in Par. 6 and in full in Part II, Procedure for Condenser-Leakage Tests, Pars. 12 to 109, inclusive. The various cases in which condenser leakage is to be determined are considered in Part III, Specific Instructions for Condenser-Leakage Tests, and it is there specified which of the methods described in Par. 6 and Part II shall be used in each case.

5 Some of the methods of conducting condenser-leakage tests can be used with salt circulating water only, and some may be used with either fresh or salt circulating water. Some require accurate apparatus and an experienced personnel, and can be used only when these are available. Some of the methods are carried on when the engine or turbine is running for the regular steam-rate tests. It is desirable to use such methods when possible because the condenser leakage may be greater when the engine or turbine is running than when it is not, due to temperature changes and vibration. Some of the methods are to be used when the engine or turbine is not running, immediately before and after the steam-rate tests. Low leakage should be shown with such methods because of possibility of increase when the engine or turbine is running.

6a *Qualitative Silver Nitrate Method.* This method is only suitable for circulating water containing an appreciable amount of salt, such as sea water or a tidewater mixture of sea and fresh water. It can be used with fresh water provided it is mixed with brine or otherwise chlorinated, such as at the suction of the circulating pump. It is the preferred method for such waters when the condenser is thought to be very tight. The test is to be made when the engine or turbine is running. The method depends upon the chemical reaction of silver nitrate with chlorides which are abundant in salt water. Samples of condensate are tested by means of a silver nitrate solution in order to determine if a trace of salt is present. If no trace is found the condenser is known to be perfectly tight. If a trace is found, the Quantitative Silver Nitrate Method, B, Pars. 33–50, must be proceeded with immediately.

6b *Quantitative Silver Nitrate Method.* This method is suitable for use with salt circulating water as specified in 6d. It is to be used when it is not certain that the condenser is tight enough for the Qualitative Method, and is the preferred method for such a case. It shall also be used when the Qualitative Method has been used and has shown some leakage. It is to be made when the engine or turbine is running. Samples of pure condensed steam (which has not passed through the condenser), of condensate from the condenser, and of circulating water are titrated against a silver nitrate solution (using potassium chromate as an indicator) in order to determine the concentration of chlorides in each. Then by calculation the amount of circulating water leaking into the steam space of the condenser is determined. This method requires some readily obtained chemical apparatus and some easily acquired skill. Complete instructions are given so that with a little effort the test may be carried out accurately.

6c *Short Direct-Weight Method.* This method is used with fresh circulating water only. It is to be used when there is agreement not to use the Standard Direct-Weight Test, D, Pars. 71–83, and when circumstances do not permit use of the Electrolytic-Conductance Method, E, Pars. 84–107. The method is used only when the engine or turbine is not running. It consists in shutting off all steam supply to the unit, running the circulating pumps at the same speed as during the steam-rate test, maintaining as nearly as possible the same vacuum in the steam space as during the steam-rate test, and weighing the amount of circulating water that leaks into the steam space as it is removed from the hotwell of the condenser. Such tests are to be made preceding and following the steam-rate test.

6d *Standard Direct-Weight Method.* This method is as above, with the addition of some refinements which make for greater precision. It is the preferred method with fresh circulating water, and is to be used unless there is definite agreement to use Methods C (Pars. 51–70) or E (Pars. 84–107).

6e *Electrolytic-Conductance Method.* This method involves a determination of the relative resistance offered to the flow of electric current between electrodes immersed in samples of pure con-

densed steam from the steam space of the condenser, of condensate from the hotwell of the condenser, and of circulating water. From these data the amount of circulating water leaking into the steam space can be calculated. The test is to be made when the engine or turbine is running. It requires a rather elaborate apparatus and an experienced personnel. The greater the amount of solids in solution in the circulating water, the easier it is to use this method.

7 With fresh circulating water, if the amount of solids in solution is appreciable and constant, and if the apparatus and personnel are appropriate, this method is preferable to the Direct-Weight Method. Extreme care must be exercised if the circulating water contains less than 20 grains of dissolved solids per gallon, equivalent to 350 parts per million. It is not possible to use this method if th circulating water contains less than half of this amount.

8 With salt circulating water, this method may be used if circumstances make it preferable to the Silver Nitrate Method, as discussed in Part III, Par. 120.

9 *Repair of Condensers.* If leakage is found to exist, the following suggestions may be helpful in eliminating it.

10 *Cold-Water Pressure Method of Locating Condenser Leaks.* A common method is to open or remove the outer heads of the condenser and to fill the entire steam space of the condenser with water. To obtain a pressure sufficient to show the small leaks which must be eliminated in order to meet the present code, there should be a head of approximately 20 ft. above the center line of the condenser. In order to have such a head, the turbine must be filled with water. All shaft seals must be packed with putty or packing of some kind, and condenser relief valves, automatic valves, etc. must be fastened down. This method, while often used, is very difficult and is not recommended, since much better results have been obtained by the method described in the following paragraph.

11 *Vacuum Method of Locating Condenser Leaks.* Open or remove the inlet and outlet water boxes. Plug one end of all of the condenser tubes with tapered, softwood plugs, pushed in by hand. Then start the air pump so as to obtain at least 18 in. vacuum. It may be necessary, in order to secure this amount of vacuum, to seal the shafts with putty, with packing, or with cold water. It is also possible to seal the shafts with the regular steam seal. However, the admission of steam will often make it difficult to maintain the desired amount of vacuum. When a satisfactory vacuum is obtained, a small gas flame or a lighted candle is to be applied to the open end of each of the tubes and to the entire tube sheet. Even a small leak will be indicated when the flame is pulled into the tube, tube sheet, or packing. The more usual types of leaks are as follows: cracked tube, corroded tube, fine holes in tube, and leaky packing.

PART II—PROCEDURE FOR CONDENSER-LEAKAGE TESTS

COLLECTION OF SAMPLES

12 In the Silver Nitrate and Electrolytic Conductance Methods, samples of condensate, circulating water, and pure condensed steam are required, and the general procedure for collection is here given for reference.

13 *Collection of the Samples* from the three different sources shall be simultaneous or nearly so. This is especially important if the circulating water is tidewater with variable salt content. The samples shall be collected in clean flasks or rubber-stoppered bottles. The pipe from which a sample is collected shall be thoroughly cleaned by a preliminary flow, the bottle or flask shall then be thoroughly rinsed with water the same as that with which it is to be filled, and finally the sample shall be collected.

14 *Samples of Condensate* shall be obtained from a cock in the condensate-pump discharge pipe or from the flow into the measuring tanks.

15 When sampling the condensate during the conduct of a steam-rate test, if the quantity of condensate drawn off in rinsing and sampling is appreciable, it shall be caught in a bucket and weighed, and the weight added to the weight of condensate used in computing the steam rate.

16 *Samples of Circulating Water* shall be collected from a cock at any convenient point.

17 Of course, a point must be selected where the pressure is above that of the atmosphere, and not in a siphon or suction line, or else a vacuum connection must be attached to the sampling vessel.

18 *Samples of Pure Condensed Steam* shall always be collected in a way which gives a representative sample of the steam entering the condenser unaffected by condenser leakage or other contaminating factors.

19 The sample of condensed steam may be obtained from the main steam line just preceding the turbine, or preferably from a turbine stage—one with pressure just above that of the atmosphere. Such a sample may be collected by a long drip pipe or by an air-cooled condenser. This may be about twenty feet or more of pipe, arranged in any convenient form of coil or bank. This coil shall be thoroughly blown out with steam before the beginning of the test. Often it is desirable to have the coil tinned on the inside. However, tin melts at 450 deg. fahr. and cannot be used with high-temperature steam.

20 With such an air condenser, a continuous flow of condensed steam shall be maintained during the test. The condensed steam should be collected in an annealed-glass or other flask which will not break due to the high temperature.

21 When the condensed steam is collected from the steam main and in cases where water is injected into the steam to maintain a given temperature, or if the steam contains a comparatively large percentage of salt, the condensed steam shall be taken from the center of the main in such a manner that the drops of water condensing on the side of the main and containing solids, cannot enter the drip pipe. This is for the reason that salt is much more readily carried by water than steam.

22 Instead of collecting the condensed steam from the steam main or from a turbine stage, it may be collected from the interior of the condenser as is required for the Electrolytic Conductance Method, Pars. 93 and 94. Collection in this way is not usually necessary for the Silver Nitrate Method, however.

23 *Preliminary Samples* shall always be taken preceding the regular steam-rate test at such intervals as seem convenient. Preliminary results shall be obtained by the same methods as are used for the regular samples, and shall be recorded on the same form of log sheet.

24 In cases where the condenser has been leaking and has been repaired or where it has not been in use for a period exceeding twelve hours, samples shall be considered as preliminary until about two hours have elapsed from the time the turbine is at test load. The reason for this is that the condenser collects solids while idle, and the first samples indicate a leakage greater than the actual. Sometimes it may be desirable to run the condenser for short periods at different loads so that various steam currents will wash all parts of the condenser. The dropping of the leakage to a reasonably constant or a negligible value will indicate when the regular test may be started. Of course if the preliminary samples indicate a negligible leakage, the regular test need not be held up for the specified two hours because of condenser leakage.

25 *Regular Samples* shall be taken at convenient intervals, at least every 30 minutes, during the entire progress of the regular steam-rate tests. There shall be at least three such sets of samples for each test point. Deductions from the steam flow for condenser leakage shall be made on the basis of the average of all of the values for a test point shown by the regular samples.

26 If the solid content of the circulating water varies, as with tidewater, it may be desirable to take samples oftener than every 30 minutes.

A—QUALITATIVE SILVER NITRATE METHOD

27 This method may be used with salt or brackish circulating water, as specified in Par. 6*a*, to which reference shall be made.

28 A bottle of silver nitrate solution shall be provided. This solution is prepared by dissolving about 1 gram of c.p. $AgNO_3$ in about 2000 cc. (2 liters) of distilled water. Exact measurement is not required.

29 *Preliminary and Regular Samples* of condensate shall be collected as specified in Pars. 9, 10, 11, 12, 13, 14, 23, 24, 25, and 26. Samples of pure condensed steam are not required. It must be certain that the circulating water contains salt, and samples of circulating water need be taken only in order to be sure of this throughout the test.

30 If the salt content of the circulating water ever becomes negligible the Silver Nitrate Method cannot be used. If there is any doubt of this, the circulating water shall be tested for salt by mixing 1 cc. of circulating water with about 100 cc. of condensate and using the method for testing the condensate described in the next two paragraphs. The Silver Nitrate Test may or may not be used, according as there are or are not perceptible white clouds.

31 *Conduct of Qualitative Silver Nitrate Test.* A few drops of the silver nitrate solution shall be dropped into each sample of condensate. The slightest trace of salt will be shown by a white cloud which will follow each drop of silver nitrate solution as it falls in the

sample. If such clouds occur, or if the final mixture is whitish, there is leakage, and either the steam-rate test shall be discontinued and the condenser repaired, or the Quantitative Method B, Pars. 33 to 50, inclusive, shall be started at once and continued throughout the steam-rate test. A log sheet shall be kept even when no leakage is shown.

32 This test is very sensitive and if there are no white clouds or perceptible traces of white in the final mixture, it may be certain there is no leakage. A tight condenser will give such a negative result. Sometimes there may be a slight trace of blueness due to negligible traces of leakage, which may be disregarded. Of course, if there is any doubt as to whether there is evidence of leakage, it must be assumed that there is leakage, and the quantitative test started. If this method is used with fresh circulating water mixed with brine or otherwise chlorinated, it must be tested frequently as specified in Par. 30 and must show perceptible white clouds.

B—Quantitative Silver Nitrate Method

33 This method may be used with salt circulating water, as specified in Par. 6*b*, to which reference shall be made.

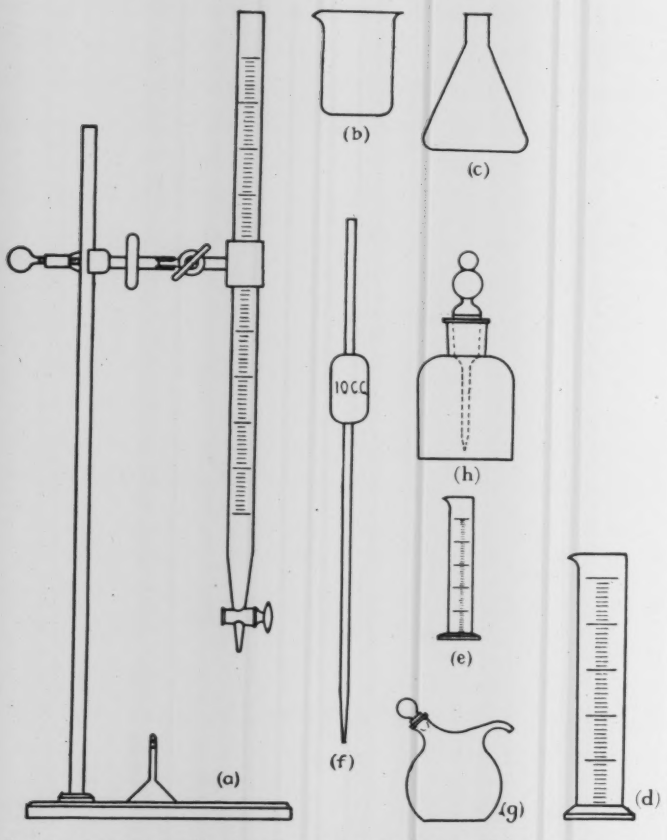


FIG. 1 APPARATUS USED IN QUANTITATIVE SILVER NITRATE METHOD

34 If this method is used with fresh circulating water mixed with brine or otherwise chlorinated, the mixing shall be done so that the various values of C_w taken throughout the test are constant within 10 per cent. The method given beyond uses K_2CrO_4 as an indicator. With an experienced personnel an alternate method is possible, using KCNS.

35 *Apparatus* required for the test comprises a burette graduated to tenths of a cubic centimeter and with a total capacity of at least 25 cc., with stand, filling funnel, etc. (Fig. 1a); a beaker (Fig. 1b) and Erlenmeyer flask (Fig. 1c) for the sample being titrated or else a 150-cc. white porcelain casserole as is preferred by some; a 100-cc. cylinder graduated to 1 cc. (Fig. 1d); a small graduated cylinder with which 10 cc. can be measured accurately (Fig. 1e), or a 10-cc. pipette (Fig. 1f); glass-stoppered reagent bottles for silver nitrate solution, and for potassium chromate indicator (a standard indicator bottle, Fig. 1g or 1h, will be found convenient); 3 rubber-stoppered bottles or flasks for samples being tested, a 1-liter annealed-glass flask to collect the condensed steam, and a gallon bottle of distilled water.

36 *Water* used for rinsing and diluting in this work shall be pure distilled water. This distilled water may be obtained from an outside source, or if it is properly salt-free, from the source of the condensed steam, Pars. 18–22. In any case the water used for rinsing and diluting shall be titrated according to the method given in Par. 40. If the silver nitrate used exceeds 5 per cent of the amount necessary for a sample of the circulating water as prepared for test (see Par. 47), the water tested is not sufficiently salt-free to be used for rinsing and diluting in this work and shall be replaced by a better supply. It may be noted that a small amount of silver nitrate is consumed by the indicator itself to give the permanent color change.

37 If the distilled water is being obtained from the source of the condensed steam each collection of it shall be titrated, and if the salt content is within the limit given above it may be added to the supply of distilled water.

38 *Solutions* used shall be prepared in advance. The volume of standard silver nitrate solution required for twenty tests (one day) is about one-half liter. The volume of potassium chromate solution required for the same tests is about 20 cubic centimeters.

39 The solutions used should be as dilute as is consistent with a sharp end point in the titration. It may be noted that the strength of solution does not affect the accuracy of the results when each set of samples is titrated with solutions of the same strength. It will be found most convenient, however, to use solutions of a uniform strength throughout. A solution containing 4.8 grams of c.p. silver nitrate per liter of distilled water will be taken as standard. One cubic centimeter of this solution is equivalent to one milligram of chlorine, to 1.66 milligrams of sodium chloride, and to about 1.85 milligrams of total salts in sea water. As an indicator, a 10 per cent solution of c.p. potassium chromate is satisfactory. This will be in the proportion of 60 grams of K_2CrO_4 to 1000 grams of distilled water. Sodium chromate may be used if desired.

40 *General Method of Procedure for Titration of Samples.* At various stages of the test, as explained later, samples must be titrated, and in each case the following method is to be used. The titration must be conducted by a man who has had some practice. This, however, is easily obtained by running a few titrations before the regular test. The titration method is as follows:

a Rinse the 100-cc. graduate and Erlenmeyer flask thoroughly with pure distilled water. Measure out 100 cc. (unless otherwise specified) of the sample to be tested in the 100-cc. graduate. Pour this 100-cc. sample into the Erlenmeyer flask or casserole.

b Rinse the burette and arrange it on the stand as shown in Fig. 1a. Fill the burette with silver nitrate solution to the zero graduation (the zero is at the top).

c Place a few drops (exactly the same number in every titration) of the potassium chromate indicator into the flask containing 100 cc. of the sample to be tested. This will give the solution a light orange color. Whirl the solution in the flask a bit to make the color uniform. Note the reading of the burette on the log sheet and then place the flask under the burette. Allow the silver nitrate solution to drip into the sample to be tested. The spout shall be in the center of the mouth so that the silver nitrate will not run down the side of the flask. When the drops begin to turn red, the flow of silver nitrate shall be stopped and the sample thoroughly shaken. Unless too much silver nitrate has been added, the red color in parts of the sample will disappear when the flask is shaken. Silver nitrate shall then be added a few drops at a time until a permanent red color remains. The base of the stand on which the flask rests should be clear white to assist the observation of color, or a white background should be secured in some similar way if a porcelain dish has not been used. This permanent red color gives the end point of the titration and the reading of the burette shall then be noted on the log sheet. The difference between the two readings will be the volume of silver nitrate required for the sample being tested.

41 The reaction between silver nitrate and sodium chloride,

$$AgNO_3 + NaCl = NaNO_3 + AgCl$$

yields white compounds which do not affect the color of the sample, which is a light orange, due to the potassium chromate indicator. As soon as all of the sodium chloride (together with other chlorides which may be present) is consumed, however, a very slight excess of silver nitrate reacts with the potassium chromate to form silver chromate a dark red compound:

$$2AgNO_3 + K_2CrO_4 = 2KNO_3 + Ag_2CrO_4$$

The end point of the titration is clearly indicated by the sharp change of color.

42 If there be a possibility that sodium hydroxide or sodium sulphate, or both, have been carried over from the boiler, the solution to be titrated must previously be made neutral as described below, to avoid white clouds due to silver hydrate or silver sulphate.

43 Add two drops of phenolphthalein indicator. If the solution colors red, add a very dilute solution of sulphuric acid (N/50) until the red color just disappears and then three drops in excess. If, upon the addition of the phenolphthalein, the solution does not color, add a very dilute solution of sodium carbonate (N/50) until the solution becomes slightly red, and then add the dilute acid until the color disappears, and then three drops in excess. By this procedure the chloride determination would be made in a neutral solution and none of the silver nitrate would be used up in forming silver hydrate or silver sulphate.

44 *Preliminary and Regular Samples* of condensate, circulating water, and pure condensed steam shall be collected as specified in Pars. 12 to 26, inclusive.

45 *Conduct of Quantitative Silver Nitrate Test.* After the preliminaries already given are arranged, the test is made as follows,

with each set of three samples. The results shall be recorded as in the sample log sheet, Par. 50. Solutions from the same silver nitrate bottle shall be used for each set of samples: First titrate 100 cc. of the condensate sample as per Par. 40. The 100-cc. sample of condensed steam is titrated next in the same manner as the condensate. If this requires the same amount of reagent as the condensate, it is not necessary to test the sample of circulating water.

46 This is for the reason that the equality of salt content indicates no addition of salt from the circulating water and hence no condenser leakage. It must of course be certain that the circulating water contains salt. If the salt content of the circulating water ever becomes negligible, the Silver Nitrate Method cannot be used. If there be any question, the test given in Par. 30 shall be used.

47 If, however, the circulating water is to be tested, it shall be diluted before titration. This is done by mixing together, in the 100-cc. graduated cylinder, 10 cc. of the circulating water sample, and 90 cc. of the *salt-free* distilled water of Par. 36. These are mixed by inverting the cylinder several times after plugging it with a clean rubber stopper. From the thoroughly mixed dilute sample, 10 cc. are measured and mixed with 90 cc. of distilled water. This 1 per cent solution of circulating water is titrated in exactly the same manner as the sample of 100 cc. of condensate.

48 *Results* shall be computed as follows:

Let c_h = volume of silver nitrate required for 100 cc. of condensate
c_w = volume of silver nitrate required for the 10 cc. of the mixture of 10 cc. of circulating water and the 90 cc. of salt-free distilled water
c_s = volume of silver nitrate required for the 100-cc. sample of condensed steam.

Then the percentage of circulating water mixed with the condensate, that is, the percentage leakage, is

$$L = \frac{C_h - C_s}{C_w} \text{ per cent}$$

For example, if $C_h = 2$ cc., $C_s = 0.4$ cc., and $C_w = 16$ cc.,

$$L = \frac{2.0 - 0.4}{16} = \frac{1.6}{16} = 0.1 \text{ per cent leakage}$$

49 If the leakage is found to be negligible, the Qualitative Method, A, may be used as long as the negligible leakage continues.

50 *Sample Log Sheet.*

CONDENSER LEAKAGE—QUANTITATIVE SILVER NITRATE TEST 1000-KW. TURBINE NO. 642

Time Dec. 22, 1922 A.M.	Condensate			Condensed Steam			Circulating Water			
	Burette reading		Amount of reagent consumed net	Burette reading		Amount of reagent consumed net	Burette reading		Amount of reagent consumed net	Per cent leakage
	Before	After		Before	After		Before	After		
			C_h			C_s			C_w	L
	cc.	cc.	cc.	cc.	cc.	cc	cc	cc.	cc	
8:30	18.3	19.8	1.5	20.0	20.3	0.3	5.1	22.1	17.0	0.07
9:00	20.1	21.4	1.3	20.5	20.7	0.2	6.3	22.8	16.5	0.07
9:30	18.7	19.7	1.0	20.0	20.4	0.4	6.7	23.0	16.3	0.04
10:00	23.4	24.3	0.9	23.0	23.4	0.4	2.3	19.7	17.4	0.03
10:30	20.2	20.9	0.7	19.7	20.1	0.4	5.8	24.3	18.5	0.02
11:00	2.5	3.1	0.6	24.3	24.7	0.4	3.7	21.1	17.3	0.01

C—SHORT DIRECT-WEIGHT METHOD

51 This method may be used only with fresh circulating water, and under the conditions specified in Par. 6c, to which reference shall be made.

52 The run made after the steam-rate tests may show 50 per cent greater leakage than specified in the Test Code for Steam Turbines or Reciprocating Steam Engines for the run before the steam-rate tests. If there is greater leakage than this after the steam-rate tests, they shall be discarded, and repeated after the condenser has been repaired.

53 *Time of Direct-Weight Test.* The tests shall be made immediately before and after the regular steam-rate test.

54 If any appreciable time is allowed to elapse, the condition of the condenser may change.

55 If there is a series of steam-rate tests, as for instance, when tests are made at different loads, the condenser-leakage tests may be made before the first and after the last one. However, it is also advisable to have intermediate condenser-leakage tests in such a case.

56 *Preparation for Direct-Weight Test.* All possibility of entrance of steam, water, or condensate into the condenser during the condenser leakage test shall be eliminated.

57 Usually the turbine or engine, with its steam piping, is connected directly to the condenser. In such event each steam inlet to the turbine or engine must be disconnected, or shown to admit no steam by use of two valves with a drip or bleeder between them. During the leakage test there is usually vacuum on the turbine and, with leaky valves, steam or water might be sucked in and give apparent condenser leakage. This may be avoided by the use of a large bleeder having a diameter about 10 per cent of the steam-pipe diameter; or by making certain that there is no suction on the bleeder by feeling it or noting if a tube connected to the bleeder will suck water from a vessel.

58 If the turbine has openings for steam extraction or bleeding for heating feedwater or for operation with low-pressure steam, or in fact openings of any kind, the above-mentioned precautions must be taken or else this method of test shall not be used.

59 No steam or water shall be used on seals of shaft glands, valves, or joints, except as specified in Par. 67. Openings usually so sealed may be closed with rubber tubing, putty, or the like. All other openings through which air might enter the condenser must of course be carefully sealed. The vacuum must be as nearly as possible the same during the condenser-leakage test as during the steam-rate test. If there is a valve between turbine or engine and the condenser, it shall be closed in order to minimize air leakage. Preparation shall be made for collecting the leakage according to one of the methods *a*, *b*, or *c*, Pars. 65 to 68, inclusive.

60 During the condenser-leakage test there may be vacuum on parts not normally under vacuum, so that there is more chance for air leakage. Painting the outside of the engine or turbine, with shellac or asphaltum with vacuum on, is often necessary in order to secure good vacuum.

61 Details of further preparations are given in Pars. 71 to 76, inclusive, to be followed if the Standard Method is being used.

62 A dry air pump is understood to mean a pump which takes nothing but air and saturated steam from the condenser, whether or not water is used in the pump itself.

63 A wet air pump is understood to mean a pump which takes both water and air from the condenser. During these tests it must be impossible for water to enter a wet air pump from any source except the condenser itself, except as specified in Par. 67.

64 *Conduct of Direct-Weight Test.* During this test the circulating and air pumps are operated. The circulating pump shall be run at the same rate as during the regular steam-rate tests. If there is a pump for condensate only, it shall be run as directed below. An air pump, wet or dry, shall be run as directed below, at such rate as will give a vacuum as nearly as possible the same as during the steam-rate tests. During the test, some circulating water may leak into the steam space, all of which shall be collected and weighed by one of the following methods, *a*, *b*, or *c*. Observation shall be taken every 10 minutes of the condenser vacuum, preferably by the same gage as used in the regular steam-rate tests.

65 (*a*) *Collection of Leakage in Steam-Rate Measuring Tanks When There Is a Dry Air Pump.* This method may be used if there is a pump for condensate only, which will handle, with steadiness, the small flow occurring during the leakage test. There shall be a gage glass on the condensate-pump inlet pipe. This method can be used only if the condensate pump keeps the glass empty or can be throttled so as to keep it filled to a constant height at all times during the leakage test. The dry air and condensate pumps shall be run throughout the test. The condensate pump of course must be properly sealed. If external sealing water is needed the method of Par. 67 may be used. There must be no gland leakage.

66 The discharge shall be measured in the regular tanks in the same manner as specified for the flow during the steam-rate tests, if this method gives results accurate within 5 per cent of the small flow involved. Otherwise a special weighing tank and scales shall be provided. The test shall continue until the leakage has maintained a value which is steady within 10 per cent for a period of at least 30 minutes. The run before the steam-rate test may reach such steady conditions in a short time. The run after the steam-rate test shall proceed as soon as the turbine is stopped, without shutting down the air, circulating, or condensate pumps. The specified steady conditions may not then be reached for several hours.

67 (*b*) *Collection of Leakage when there is a Wet Air Pump.* The circulating and wet air pumps shall be run and the wet-air-pump discharge measured in the regular condensate-measuring tanks in the same manner as specified for the flow during the steam-rate tests, if possible. Further de-

tails shall be as in Par. 66 above. This is the only method possible with a wet air pump. If the condenser leakage is so small that the wet air pump cannot maintain an appreciable vacuum because its valves are not sealed, or the like, the following device shall be used. A previously weighed tank of water is to be provided, into which the wet air pump shall discharge during the leakage test. From this tank a suction pipe shall be run into the steam space of the condenser. The weighed water will then circulate round and round and furnish the proper seal. When the test is shut down all of the weighed water shall be known to drain back into the tank. The condenser leakage is then given by increase in weight.

68 (*c*) *Collection of Leakage in the Condenser When There Is a Dry Air Pump.* The condensate pump is not to be run and the pump inlet or outlet shall be known to be closed. A drain shall be provided in the piping between the hotwell and the pump, which shall be opened and closed as directed below. Before the leakage test, water must be introduced through the drain opening, by means of a hose or funnel, so as to completely fill the entire piping and condensate pump below the drain opening. There must be no leaks in the condensate pump glands, or elsewhere. A vessel shall be provided for collecting the condenser leakage which will flow out when the drain is opened. A vent on the condenser must also be provided which can be opened when the drain is opened. If the Standard Direct Weight Method D is being used, a preliminary run shall be made before the regular run, as directed in Par. 77. The regular run is started at a time which is to be noted, by closing the drain and vent and starting the dry air pump. The run is to be continued for an integral number of hours, from one to four, depending upon the importance of the test and the rate of leakage. The run is finished by stopping the dry air pump and opening the drain and vent. The leakage for the period of the test will then flow into the vessel provided.

69 Throughout the test, by either method, the temperature of the condenser shell must be lower than the boiling point corresponding to the condenser absolute pressure, in order to prevent loss by evaporation.

70 *Calculation of Results for Short Direct-Weight Method.* From the time and the weight of the condenser leakage, the leakage per hour shall be found. The values before and after the steam-rate tests shall be seen to be within the limits given in Pars. 51 and 52. The average of the values before and after the steam-rate tests gives the leakage per hour, which must be subtracted from the flow per hour measured during the steam-rate test.

D—Standard Direct-Weight Method

71 This method may be used only with fresh circulating water, and under the conditions specified in Par. 6*d*, to which reference shall be made.

72 When this method is used, all of the directions for the Short Test C, Pars. 53 to 69 inclusive, shall be followed with the following additional details.

73 *Differential-Pressure Observations.* When the condenser leakage test is ready to be started, the air pump shall be run for a short time and an observation made of the vacuum which will occur during the leakage test. If the vacuum is not within 1 in. Hg of the value expected during the regular steam-rate tests, additional efforts shall be made, by better sealing, attention to air pump, etc. to secure a better vacuum. After this has been done, if the vacuum is not within 2 in. Hg of the value expected during the steam-rate tests, and if it is not positively known that the pressure of the circulating water during the leakage test is within 2 in. Hg of the pressure during the steam-rate test, gages shall be used to give the circulating-water pressure during both the condenser leakage test and the steam-rate tests. These shall be two mercury columns or gages of any approved kind connected to the water space of the condenser near the circulating-water inlet and outlet, respectively. They shall give either the direct pressures of the circulating water or the differential pressures between the condenser vacuum and the circulating water inlet and outlet, respectively.

74 If direct gages are used, the differential pressures shall be computed, careful account being taken of the pressures above and below atmospheric pressure. If the differential pressures are taken, the instructions given in Part 5, Pressure Pipes and Holes, shall be followed so as to insure that one pressure pipe is filled with water and the other with gas; and the proper allowance shall be made for the hydrostatic head on one side.

75 These differential pressures shall be seen to be at least half of the values expected during the steam-rate test. If either is less than half, or if it is in the opposite sense so that the leakage is reversed, the test cannot be continued until a better vacuum is obtained during the leakage test, by better sealing or work on the air pump.

76 If there is a poor vacuum, and if the siphon effect on the circulating water is great, it may happen that the absolute pressure in the steam space is greater than the absolute pressure at the water outlet, which gives a differential pressure in the opposite sense from that occurring during the steam-rate test, so that the direction of leakage is reversed and the leakage test is absurd.

77 *Preliminary Run.* When the leakage test is to be made by the Standard Method, and when method C, Par. 68, is used to collect the leakage, then before the regular condenser leakage run, there is to be made a run for a preliminary period of 10 to 30 minutes, in order to drain the condenser, to establish steady conditions, fill water pockets, etc. The circulating pump is of course to be kept running. The procedure is as follows:

- *a* Close hotwell drain and vent
- *b* Start air pump
- *c* Run for 10 to 30 minutes
- *d* Stop air pump
- *e* Open drain and vent, and let leakage run off
- *f* After the first flow there will be a trickle from the drain, which may gradually decrease for a long period or may settle down to a small steady flow. After a time the rate of flow should be found by collecting it for 5 to 10 minutes in a small vessel.

78 If the rate of flow is not visibly decreasing, the amount collected will show either that the condenser is obviously leaking so much that it must be repaired before proceeding further, or that it is desirable to make the regular leakage run in order to obtain the actual amount of leakage.

79 If the rate of flow is visibly decreasing, it should be measured successively until it becomes constant or becomes less than the allowable limits, and the regular leakage run should then be conducted as per Par. 47.

80 During the regular leakage run, readings shall be taken every 10 minutes showing the condenser vacuum and the differential pressure between the steam space and the water space near inlet and outlet, respectively, as per Pars. 73 to 76.

81 *Calculation of Results of Standard Direct-Weight Method.*

L = corrected value of condenser leakage, lb. per hr.

p^r_1 and p^r_2 = observed values in any convenient units of differential pressures between steam space and water spaces at condenser inlet and outlet, during steam-rate tests

p^l_1 and p^l_2 = ditto, during condenser leakage test, using the same units as above

L_0 = observed condenser leakage, lb. per hr., obtained by dividing the actual leakage in pounds (taking account of tare) by the number of hours of the leakage run.

82 The leakage is proportional to the square root of the differential pressure, and the average leakage may therefore reasonably be assumed to be proportional to the average of the square roots of the differential pressures at inlet and outlet. It follows that

$$L = L_0 \frac{\sqrt{p^r_1} + \sqrt{p^r_2}}{\sqrt{p^l_1} + \sqrt{p^l_2}}$$

83 The values of L from the leakage tests before and after the steam-rate tests must be seen to be allowable within the limits. The average of the values of L from leakage tests before and after the steam-rate test gives the leakage per hour which must be subtracted from the flow per hour measured during the steam-rate test.

E—Electrolytic Conductance Method

84 This method may be used under the circumstances specified in Par. 6*e*, to which reference shall be made.

85 *General Methods of Measurement.* The usual system of applying the Electrolytic Conductance Method to the measurement of condenser leakage involves a determination, by a proper electrical instrument, of the conductance, or resistance to flow of electric current, between electrodes immersed in samples of condensate, pure condensed steam, and circulating water. For this purpose a number of methods are available, any one of which, when properly used, will give satisfactory results. Good commercial instruments of several types are on the market, and one of these may be used, such as the Dionic water tester, distributed by Jas. G. Biddle, the Leeds and Northrup conductivity bridge, or the Esterline-Angus concentration meter. The detailed instructions of the manufacturer are, of course, to be followed, as well as the general instructions herein. An apparatus may also be especially prepared, according to instructions given by Cald-

well in *Power*, Aug. 9, 1921. It has also proposed to use an apparatus consisting of current from 110 or 220 a.c., an accurate voltmeter and two half-inch-round carbon electrodes covered with glass tubing which have been shellacked on in such a manner that on each end the carbon rods will project one inch beyond the tubing. The rods should be set parallel to one another on about one and one-half inch centers.

86 Alternating current is frequently used. Then a transformer may be used to eliminate grounds as well as to give the proper value of e.m.f. It should be inserted in the supply circuit in such a way as to isolate the test circuit from the supply circuit. Of course the voltage must be constant.

87 Direct current is also used. It will produce an evolution and deposit of gases on the electrodes and result in an apparent increase in resistance. There will also be a back e.m.f. of polarization. If the current density at the electrodes is low and the path through the electrolyte is long and of rather small cross-section, the errors due to increased resistance on account of the collection of gases at the electrodes and the back e.m.f. of polarization may be reduced to a minimum. Polarity may also be reversed and a mean reading used.

88 The Dionic water tester uses direct current generated by a "megger" or hand-operated dynamo and the calibration of the instrument takes care of the back e.m.f. of polarization. In the Dionic water tester the conductance is given directly in units of conductivity (specific conductance), the unit being the reciprocal of one megohm per centimeter cube between parallel faces.

89 *Readings* of the conductance, resistance, or current flow of the various samples may be made on any convenient scale, which need not be calibrated. The readings may be expressed simply as "units." It is, of course, important that all readings be in terms of exactly the same unit.

90 *An Electrode Container* in the form of a tube or otherwise shall be provided, within which samples of condensed steam, circulating water, or condensate may be placed as required or several independent electrode outfits may be used, permanently installed in different pipe lines. In the latter case, each of the electrode outfits must temporarily be placed in an external container and found to read alike on the same sample. The electrode container shall be carefully insulated from grounds.

91 *Insulation of the Electrodes* from each other shall be well provided. When immersed in the sample, the relation of the electrodes to the sides and bottom of the container shall be such that a movement of the container will not affect the resistance by changing the path for current flow between electrodes.

92 *Samples* of condensate, circulating water, and pure condensed steam shall be collected as specified in Pars. 12 to 26, inclusive. The sample of condensed steam may be collected from the steam main or preferably from a turbine stage, as per Pars. 19, 20 and 21 when there is salt circulating water.

93 Samples of condensed steam shall be collected beyond the turbine low-pressure end as described in Par. 94 when there is fresh circulating water, and may be so collected when there is salt circulating water. This is for the reason that dissolved solids may be deposited by the steam in the turbine or the dissolved gases may vary, and this may change the conductivity of the condensed steam so as to appreciably affect the results.

94 The condensed steam beyond the turbine may be collected from the exhaust space of the turbine beyond the last wheel, or from any part of the condenser shell where it can be certain that leakage from the tube plates or tubes cannot affect the result. The possibility of a split tube shall be considered. An air-cooled condenser may be used as described in Par. 13 or a glass tube cooled by water or ice, or the sample may be taken by means of a funnel arranged within the condenser so as to catch condensed steam. The collecting tube may be lead into a flask closed by a rubber stopper with two holes, the second opening being connected to the dry-air-pump inlet pipe. The condensed steam will then flow into the flask by gravity.

95 The routine prescribed requires that samples be collected regularly as the steam-rate tests proceed. This is especially important with the condensed-steam sample because its conductance depends upon the solids in the steam leaving the boiler, which may vary with the load.

96 *Precautions for Collection of Samples* in addition to those of Pars. 6 to 26 are as follows:

Metal pipes and valves or glass tubes connected by very short lengths of high-grade rubber tubing shall be used for collecting. Ordinary rubber tubing contains filler and sulphur, and shall not be used. Even pure-rubber tubing shall not have appreciable surface exposed to the sample being collected. Pipes must of course be thoroughly washed out by preliminary flow which is rejected before collecting samples.

97 Unnecessary exposure of the samples to the atmosphere must be prevented to avoid the change of conductance which occurs due even to very slight solution of gases, etc., from the atmosphere or the giving off of previously dissolved gases. For this reason it is necessary to have the upper part of a vessel filled with the vapor of the sample whenever possible. Care must also be exercised to keep the samples free from dust, dirt or other foreign particles, as such material will affect the conductance.

98 *Care of Electrodes.* Always be certain that the electrodes are clean and free from deposits. A hot solution of 10 per cent muriatic acid in water will usually remove scale. Washing in benzol will remove oil.

99 If small platinum electrodes are used with a Wheatstone bridge, it is advisable to platinize them before use. This operation is carried out by immersing the electrodes in a 3 per cent solution of platinic chloride with 1/40 per cent lead acetate added, and passing a small battery current through the cell to deposit the platinum black. By reversing the current both electrodes may be platinized. The function of this platinizing is to increase effective surface area and eliminate the formation of gas films.

100 Always wash the electrodes well after cleaning or platinizing, the final wash being made in distilled water or in the next sample. Be particularly careful that the position of the electrodes is not altered to the slightest degree.

101 Before using the electrodes it is a good plan to measure their resistance or current flow with the electrodes out of the water under test. They should first be shaken or gently jarred to dislodge water that may remain due to surface tension. A very high resistance and zero current flow should be secured. If not, the reading indicates the presence of a circuit across wet lead wires of electrode terminals.

102 *Electrodes Shall Be Washed*, as well as their container, whenever change is made from one sample to another, in the purest distilled water available—which may be part of the condensed-steam sample—or in the next sample. Again be careful not to alter position of electrodes.

103 *Uniform Temperature* of all samples shall be maintained while conductances are being measured. A convenient standard temperature shall be selected in advance, and every effort shall be made to make all tests at this temperature, as shown by a single thermometer. The difference between the selected standard temperature and that of any sample in test shall not exceed 5 deg. fahr. It is however, highly desirable to make the difference much smaller than this. 68 deg. fahr. (20 deg. cent.) is sometimes used as the standard temperature, but this value need not be used if another is more convenient.

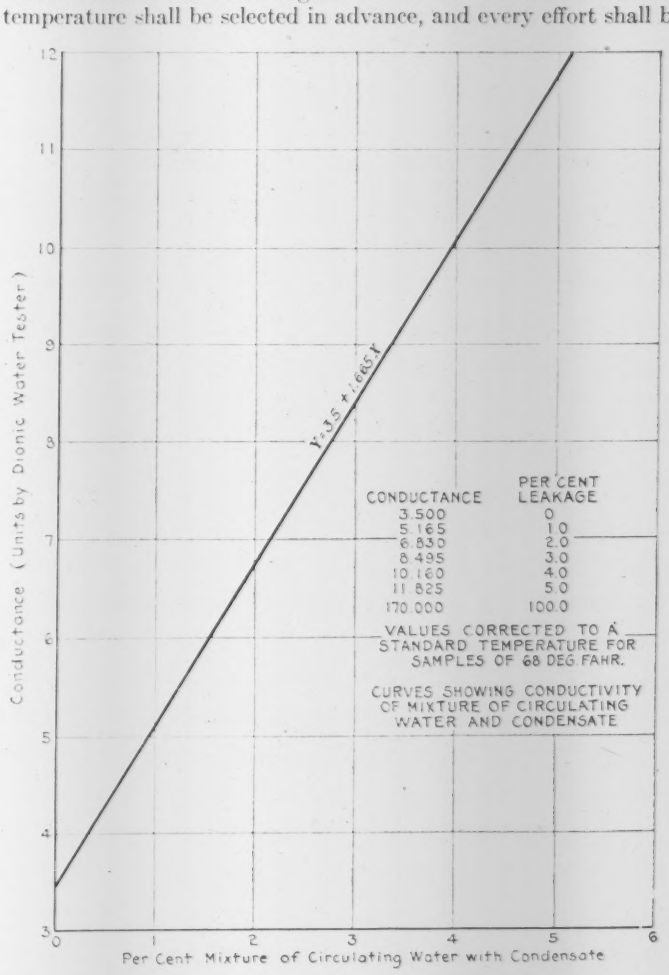

Conductance (Units by Dionic Water Tester)
Per Cent Mixture of Circulating Water with Condensate
Y=3.5 + 1.665 X
CONDUCTANCE
PER CENT LEAKAGE
3.500 0
5.165 1.0
6.830 2.0
8.495 3.0
10.160 4.0
11.825 5.0
170.000 100.0
VALUES CORRECTED TO A STANDARD TEMPERATURE FOR SAMPLES OF 68 DEG. FAHR.
CURVES SHOWING CONDUCTIVITY OF MIXTURE OF CIRCULATING WATER AND CONDENSATE

FIG. 2 CALIBRATION CURVE FOR APPARATUS

104 *Temperature Correction Factor.* Let t_1 deg. fahr. be the standard temperature and t_2 deg. fahr. be the temperature at which resistance or conductance is actually measured. Then the correction for unavoidable small difference from the standard temperature shall be made by multiplying the resistance as actually measured, by the empirical factor $(t_2 + 5)/(t_1 + 5)$ or by multiplying the conductance or current flow by $(t_1 + 5)/(t_2 + 5)$. A curve or alignment chart may be used for making the correction given by this formula. Such a chart has also been used for correcting for change of conductance of the circulating water (with constant conduc-

tance of condensed steam) although it is usually best to take account of this change by the method of Pars. 105–106.

105 *Calibration of Apparatus in Terms of Percentage Leakage.* After the preliminaries already given are arranged, and after a preliminary period conducted as per Par. 23, when it is evident that constancy of conductance readings is being obtained, samples of condensed steam and circulating water collected in the regular way and tested at the selected temperature shall be used as the basis for a calibration curve in the following way. If convenient, several successive samples of condensed steam and circulating water may be collected and tested and the results averaged. (As specified, electrodes and container shall be washed after each test, and the selected temperature shall always be maintained.)

106 Add 1 cc. of a sample of circulating water to a liter of a sample of condensed steam and find the conductance, which will correspond to 0.1 per cent leakage. Repeat with 2 cc. of circulating water, etc., until data are obtained so that a calibration curve can be constructed like Fig. 2. The ordinates may be units of either conductance, resistance, or current flow. This curve applies to condensed steam and circulating water with the same conductance as the standard samples, at the standard temperature. If conditions during the steam-rate tests are constant, the calibration need be made for only a single mixture of condensed steam and circulating water, provided that such a mixture can be proportioned to give the same conductance as that of the condensate during the steam-rate test.

107 *Conduct of Electrolytic-Conductance Test.* After the preliminaries already given are arranged, the test is made by finding and recording the conductances (or resistances or current flows) for each set of three samples. If the conductances of the samples of condensed steam and circulating water, respectively, have not changed appreciably from the values of the calibration curve, the conductance of the condensate gives the percentage of leakage, by referring to the curve.

108 If at any time during the test the conductance of either the condensed steam or the circulating water, or both, are appreciably different from the values of the original calibration curve, a new curve shall be made as per Pars. 105 and 106, using samples taken during the time the regular samples are being taken.

109 If the circulating water contains so great an amount of dissolved salts as to give readings of conductances, resistance, or current flow off the scale of the instrument, the constancy of the conductance of the circulating water during the tests may be determined by adding to one liter of condensed steam some convenient (but constant) number of cubic centimeters of circulating water and finding the conductance of the mixture instead of the pure circulating water, throughout the tests. The samples of circulating water and condensed steam shall be taken simultaneously or nearly so. Of course, if the condensed steam has changed in quality a new curve shall be drawn, even if the quality of the mixture of circulating water and condensed steam remains the same.

PART III—SPECIFIC INSTRUCTIONS FOR CONDENSER-LEAKAGE TESTS

110 The method to be used in determining the amount of circulating water leaking into the steam space of a surface condenser depends upon the nature of the circulating water, that is, whether it is fresh water or salt water.

111 In order that the weight of condensate from a surface condenser, corrected for leakage of circulating water into the steam space, shall be considered satisfactory as a measure of the steam flow to the engine or turbine, there are given in the various test codes maximum amounts of leakage that will be permissible with the various methods of condenser-leakage measurement. These values are percentages of the rated flow at rated load, but give the amount of leakage which is permissible for steam-rate tests at any load.

112 If the leakage (before a test or during a test) exceeds the limit prescribed for the method used, the steam-rate tests shall be discontinued until the condenser has been repaired, the leakage test repeated, and the leakage found to come within the prescribed limit.

Fresh Circulating Water

113 When the circulating water is fresh the condenser-leakage test shall be made by the Standard Direct-Weight Method used before and after the steam-rate tests, unless there is agreement by all parties interested to use the Short Direct-Weight Method before and after the steam-rate tests, or the Electrolytic-Conductance Method at intervals during the steam-rate tests.

114 *Standard Direct-Weight Method.* The procedure to be followed is given in Part II, C and D, Pars. 53 to 83, inclusive.

115 *Short Direct-Weight Method.* In special cases where all parties interested agree, some of the precautions of the Standard Direct-Weight Method may be omitted. In such cases the procedure to be followed as given in Part II, C, Pars. 51 to 70, inclusive.

116 *Electrolytic-Conductance Method with Fresh Circulating Water.* When the circumstances, apparatus, and personnel are appropriate, this test can be conducted in a satisfactory way with fresh water. However, at the present time all of these points cannot always be ascertained, so that this method is to be used only when there is agreement to it by all parties interested. When these points can be ascertained, this method is preferable to the Direct-Weight Method because leakage is determined while the steam-rate tests are in progress. The leakage at that time may be greater than that shown by the Direct-Weight Method, due to temperature changes and vibration.

Salt Circulating Water

117 When the circulating water is either brackish or salt, the condenser-leakage test shall be made by use of silver nitrate according to either the Qualitative or Quantitative Methods unless there is agreement by all parties interested to use the Electrolytic-Conductance Method. The Direct-Weight Method shall not be used with salt circulating water because the Silver Nitrate Method is always possible and is much more satisfactory.

118 *Qualitative Silver Nitrate Method.* At 30-minute intervals during the entire steam-rate test the Qualitative Silver Nitrate Method may be used according to the procedure given in Part II, A, Pars. 27 to 32, inclusive, followed by the Quantitative Silver Nitrate Method, if leakage is disclosed at any time. Of course, if the Qualitative Method never shows leakage, the Quantitative Method will never be required. This will be the case with a condenser in good condition. For such a condenser this method is the preferable one.

119 *Quantitative Silver Nitrate Method.* At 30-minute intervals during the entire steam-rate test, this method may be used if the condition of the condenser does not warrant use of the Qualitative Method, or if the Qualitative Silver Nitrate Method indicates that there is leakage. This method is preferable if the condition of the condenser is uncertain or known to be bad. The procedure to be followed is given in Part II, B, Pars. 33 to 50, inclusive.

120 *Electrolytic-Conductance Method with Salt Circulating Water.* When the apparatus and personnel are appropriate, this test can be conducted in a satisfactory manner. It is much easier to use with salt water than with fresh water. However, in most cases the Silver Nitrate Method is still easier, so that this method is to be used only when there is agreement to it by all parties interested. The procedure to be followed is given in Part II, B, Pars. 84 to 109, inclusive.

121 When the Electrolytic-Conductance Method is used for salt circulating water, it may be necessary to dilute the samples of circulating water before determining their conductance, as specified in Par. 109.

While there will always be routine engineering work to be done, for which a cut-and-dried technical training may be sufficient, such jobs must necessarily be subordinate. The worth-while jobs will be those whose holders are continually wrestling with new problems. As pointed out by speakers at the recent Pacific Coast Convention of the A.I.E.E., research training is always expensive, but there are several ways in which industry can support the type of education from which it is certain to profit so highly. Among these are the establishment of numerous research fellowships in engineering colleges, coöperative work between research departments of industrial organizations and educational institutions, the establishment of research foundations as integral parts of engineering colleges, the sabbatical exchange of engineers between engineering faculties and engineering staffs of manufacturing companies, and the taking out of patents by universities on discoveries made in their laboratories.—*Power*, September, 1925, p. 494.

SURVEY OF ENGINEERING PROGRESS

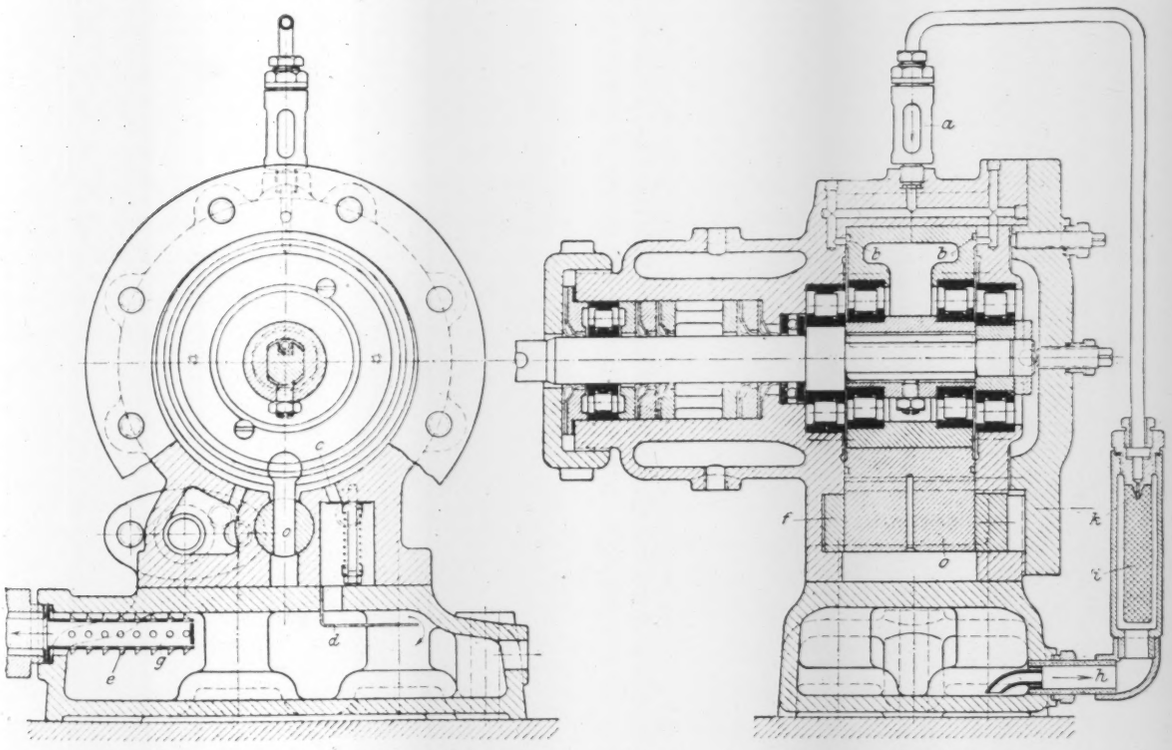
A Review of Attainment in Mechanical Engineering and Related Fields

AERONAUTICS

The Beardmore W.B.-XXVI

A TWO-SEATER fighter of unconventional design, fitted with a 375-hp. Rolls-Royce engine. In order to increase the aerodynamic efficiency of the machine, wing-bracing wires were left out and a large gap-to-chord ratio used. The flat-sided fuselage is of fairly small cross-sectional area and is not directly attached to either upper or lower plane. The biplane wings are braced by three struts on each side and attached to the fuselage by pin joints so that the wings can be quickly dismantled and rerected. No adjustment of any kind is necessary after the final truing up of the machine, the wings being rigidly locked in position merely by inserting the necessary bolts and pins.

The engine is totally enclosed and a neat nose has been made possible by employing a special type of Lamblin radiator. This is located on the leading edge of the bottom center section, a position which combines the advantage of good accessibility with very small drag. (*Flight*, vol. 18, no. 34/869, Aug. 20, 1925, pp. 534–536, *d*)



FIGS. 1 AND 2 GUETTNER HIGH-SPEED ROTARY PISTON COMPRESSOR (SCALE 1 TO 4)

(*a*, oil inlet; *b*, grooves on faces of piston; *c*, pressure piping; *d*, baffle plate; *e*, umbellar plates on pipe connection *g*; *f*, gudgeon; *h*, delivery pipe; *i*, sieve; *k*, nozzle; *o*, tongue.)

COMPRESSORS

Rotary Piston Compressors

DESCRIPTION of the Guettner small ammonia refrigerating compressor. This compressor is of the eccentric-rotor type, but is claimed to embody certain features which make it an efficient high-speed machine.

The main features are the very small degree of eccentricity of the piston as compared with the older types, and also the high speed at which the machine can be run (up to 1500 r.p.m.). Because of this high speed it is claimed that when the piston is in its highest position, direct communication between the suction and pressure sides is established for a brief period of time. At low speeds this would result in distributing periodic fluctuations of pressure, which does not occur with the high speed employed. In order to make the machine efficient a very high grade of workmanship is required. The latest type of these compressors built by Sylbe and Pondorf, of Schmölln, Thuringia, is shown in Figs. 1 and 2. The chief difference from older types consists in the fact that the most sensitive parts, gudgeon *f* and tongue *o*, have been shifted to the bottom of the machine, so that all risk of jamming owing to insufficient lubrication is thus precluded. Gudgeon *f* is now made from white (anti-friction) metal. The lubricant enters the casing from above at *a* (Fig. 2), spreads out to both sides, reaches the faces of the piston over grooves *b*, trickles through the whole machine, and finally collects at the bottom at gudgeon *f*. All superfluous oil is entrained into the pressure pipe *c* by the compressed gases. The base of the casing is designed as an oil separator and sump. For this purpose a baffle plate *d* and a pipe connection *g* with eight umbellar plates *e* are provided. The baffle action and frequent deflections cause the oil to be separated from the compressed gases. The maximum gas pressure prevailing in this chamber forces the oil into the delivery pipe *h* (Fig. 2), and after passing through sieve *i* and nozzle *k* it is returned to the compressor again at the top.

The original article gives data of tests, but only on compressors of an older type. Values of volumetric efficiency up to 85 per cent and indicated efficiencies up to 82 per cent are said to have been obtained, these efficiencies remaining fairly constant between 600 and 1000 r.p.m. It would appear that with increasing condensing temperatures, i.e., increasing back pressure, the efficiencies decrease much more slowly than is the case with compressors having reciprocating pistons. At an evaporation temperature of —10 deg. cent. and a condensing temperature of 37 deg. cent., for instance, the efficiencies were found as high as to be 71.1 (volumetric) and 71.9 (indicated) per cent, which makes this type of construction suitable for unfavorable conditions of cooling water such as occur, for instance, in tropical countries. (Prof. R. Plank, Dr. M. Krause and W. Tamm, in a report published by the Machine Laboratory of the Dantzic Technical University; abstracted through *Engineering Progress*, vol. 6, no. 8, Aug., 1925, pp. 257–261,6 figs., *d*. Compare *Zeitschrift des Vereines deutscher Ingenieure*, vol. 69, nos. 13 and 15, March 28 and April 11, 1925, pp. 393–397 and 477–481, *d*)

HYDRAULIC ENGINEERING

Hydraulic Turbines for New Zealand

DESCRIPTION of two turbines built by Gilbert Gilkes & Co., Ltd., Kendal, England, designed to develop each a maximum at 208 hp. on a net fall of 410 ft. when running at 1000 r.p.m.

The turbines are of the Turgo impulse type, somewhat similar to the Pelton type. The jet impinges on one side of an angle to the wheel and discharges on the other. The jet is formed by the water passing through a circular nozzle *A*, Fig. 3. In the center of this nozzle is a gun-metal spear *B*, which by moving in or out varies the size of the jet in the ordinary way. The spear is fixed to an Admiralty bronze rod *C*, and is, as will be observed, supported near the nozzle to insure its being kept perfectly central. The other end is supported in what Messrs. Gilkes call the economizer. The supply pipe *D*, which is connected to the pipe line, is bolted to the turbine case, and secures the nozzle holder *E* in position. Situated inside the casing is a duralumin deflector *F*, which is bolted to two levers *G* that are fixed to a vertical rocking shaft *H*. The latter is coupled by means of aluminum levers and a link to a second rocking shaft *J*, arranged outside the case. This shaft is connected by levers *K* to the governor sleeve *L*, and the governor is mounted directly on the end of the turbine shaft *M*. The two rocking shafts *H* and *J* are carried on ball bearings throughout.

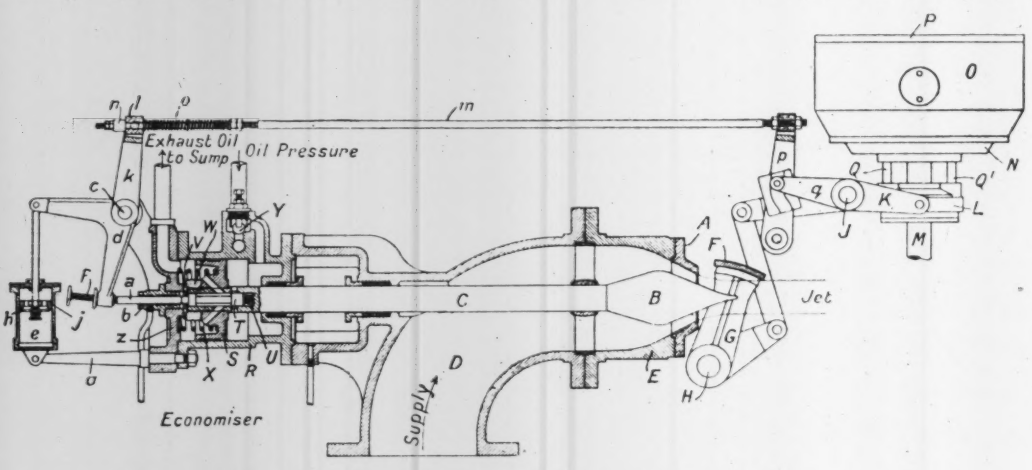


FIG. 3 TURBINE-CONTROL MECHANISM OF THE TURGO TURBINE

The governor, which is fitted to each of these turbines, is the Gilkes patented shaft governor, and is directly coupled to the deflector *F*. If the speed rises above normal the deflector is moved into the jet and diverts the water from the wheel until the speed is brought back to normal. At the present date the results of the tests of these machines have not been received, but similar governors are said to have given satisfactory results. With full load thrown off, the momentary rise in speed has been found to be between $1^1/_2$ to 2 per cent above normal light-load speed. These machines are not fitted with any flywheel.

Imagine the plant to be running on full jet when all the load is suddenly thrown off. The governor immediately moves the deflector *F* into the jet. At the same time the push rod *m* is moved in the direction to close the economizer, which has the effect of compressing the spring *o* at the end of the push rod. The compressed spring maintains a thrust on the block *l*, through which the rod slides, and working against the resistance of the dashpot *e* quietly operates the tappet *a* of the economizer valve, closing the nozzle until the required size of jet is arrived at. At this stage the spring *o* will have returned the block back to its normal position against the adjustable collar *n* on the end of the push rod. If now the load be suddenly thrown on again, the governor immediately moves the deflector away from the existing jet, and at the same time pulls the push rod. The adjustable collar being hard up against the gun-metal block, immediately moves the lever *d*, and as the dashpot is offering no resistance when it is moved in that direction, the piston valve *T* is operated quickly, thus increasing the size of the jet to that required by the load.

The makers claim that with this arrangement very rapid governing can be obtained without any danger of shock on the pipe line. In the event of the oil pump failing at any time, the governor will still have complete control over the machine if the load is thrown off. The only effect the failure can have is that if more load be put on the economizer cannot open up. Owing to leakage past the economizer ball valve the plant will gradually shut down.

A mechanism is fitted to both these turbines so that in the event of any trouble the turbine is automatically shut down. It consists of a link coupling together the levers on the governor rocking shaft and the deflector shaft. This link is made up of two circular sleeves, one sliding inside the other, each lever being connected to a sleeve. Inside these sleeves is a strong spring always tending to force the two ends apart, but normally a catch prevents that from happening. This catch is coupled up to a magnetic contactor, which is supported from the branch pipe by a light connecting rod. While current is supplied to the magnetic contactor the catch is held in position; but if the current is interrupted by any means the catch is released and the spring forces the ends of the link apart. When that happens the deflector is moved into the jet irrespective of the position of the governor sleeve, and at the same time the economizer is operated, with the result that the machine is shut down. Fixed to one sleeve is a screwed spindle, which passes through the end of the other, and a small handwheel screwed on to this spindle is used for compressing the spring and enabling the catch to be reset. Interconnected with the magnetic contactor are thermostats, which are fitted to all main bearings and generator windings so that in the event of any undue rise of temperature the plant automatically shuts down. (The *Engineer*, vol. 140, no. 3635, Aug. 28, 1925, pp. 223–224, 5 figs., *d*)

Automatic Eliminators for Alluvial Matter in Water-Turbine Supplies

ALLUVIAL matter may be quite detrimental to the water turbine, particularly as it may contain more than just sand and clay. Thus, at Rio Remac in Peru, the water carries much clinker from the copper works in addition to its natural alluvial matter which constitutes a class of material that could have a very injurious action on the turbines.

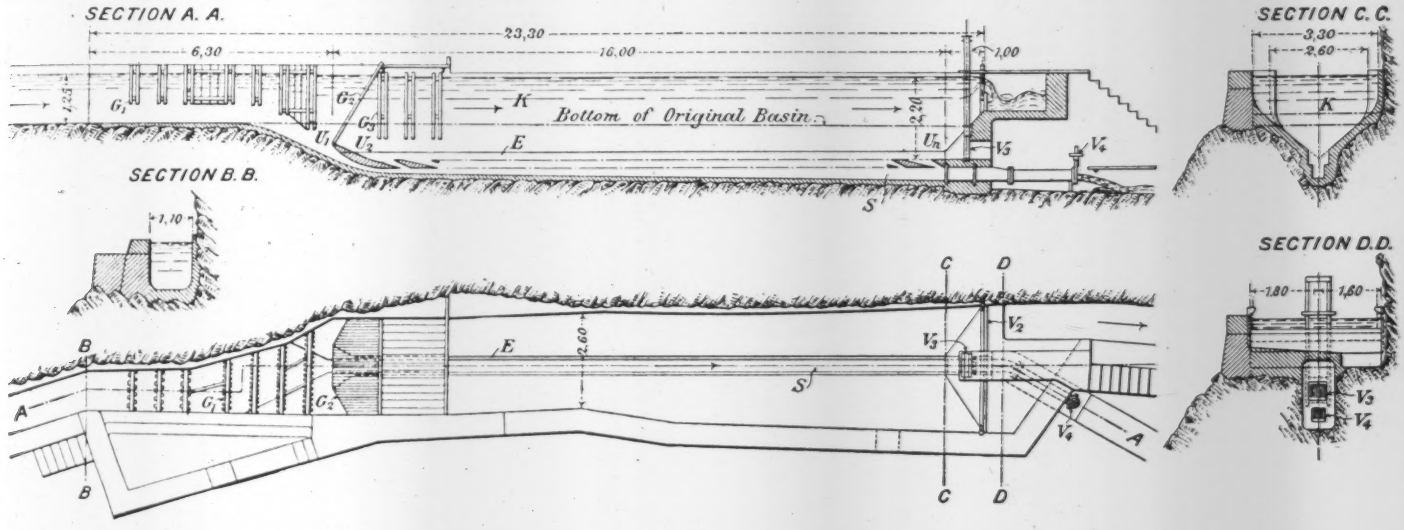
The modern practice in the elimination of such matter is illustrated by the installation at the Zermatt power station of the Gornergrat Electric Railway in Switzerland. In this station, which has a capacity of 1000 hp., at first two decantation basins were constructed one above the other, but it was found that these basins had to be cleared out as frequently as two or three times a day, resulting on each occasion in an interruption of the operation of the station, and consequently of train service, and involving a heavy item of expense in labor.

The alluvial matter transported during torrential conditions was extremely large. The automatic eliminator was constructed out of one of the existing settling basins. Its arrangement is shown in Figs. 4 to 8.

The eliminator consists of a single decanting channel *K* fitted with two sets of screens G_1 and G_2 which serve to quiet the entering water. Behind these a screen of finer mesh G_3 is situated. Along the center line of the basin floor a narrow passage *E* forming a sink has been cut. This communicates through a series of openings U_1, U_2, ... U_n with the flushing channel *S* situated below it and really forming part of the same central channel, but being a little narrower, as shown in Fig. 6. The flushing channel leads to an outlet pipe controlled by the sluice valve V_4, which can be seen in Figs. 5 and 7. The equipment also includes a weir V_2, the height of which may be regulated, and an emptying sluice V_3.

The operation of the automatic eliminator is as follows: The

water charged with alluvial matter entering the eliminator comes in contact with the screen G_1. This reduces its velocity to some extent and distributes it uniformly over the section of the eliminator K. At the same time the largest pebbles or other pieces of matter brought down by the water are stopped by the screen, and pass through the orifice U_1 direct to the entrance of the flushing channel S. The action of screen G_1 prevents blocking below the finer screen G_2, the duty of which is to trap such of the larger material as passes screen G_1, and especially large floating bodies. The screens G_2 and G_3 complete the stilling of the water, which after



FIGS. 4 TO 8 AUTOMATIC ELIMINATOR FOR ALLUVIAL MATTER AT THE ZERMATT POWER STATION, SWITZERLAND

passing G_3 flows gently down the basin, allowing the solid matter it carried to sink on to the inclined planes which form the bottom of the sink. Since the sluice valve V_4 is open and the flushing channel communicates with the basin K through the orifices U_1, U_2, etc., a certain quantity of water, depending on the height of the water in the basin, passes out through the channel S. The rate of flow of the water is sufficiently high to carry off all the deposited matter through the sluice V_4, whence it passes again into the torrent.

To fulfill their object the sink E and the flushing channel S should evacuate automatically and continuously all gravel, sand, and mud which are precipitated on the bottom of the basin K. The quantity of these materials and their distribution over the length of the canal is difficult, or impossible, to estimate in advance. As a result of model experiments and long experience, however, successful results have been obtained even with eliminators of great length. The proportions between the openings in the screen G_1 and the dimensions of the orifice U_1, and between the openings in the screen G_2 and the dimensions of the orifices $U_2 \ldots U_n$, have been so chosen that there is no probability of material which can pass the screens choking the orifices U_2, etc. It is clear that if these orifices become obstructed the eliminator would cease to operate, and the basin would quickly fill up with alluvial matter. As long as the supply of water in the torrent is greater than that required by the power station, which is the case during almost the whole of the season during which the railway is in operation, the sluice V_4 is open. Apart from the cleaning required by the screen G_2, the plant requires no attention and continues to operate automatically without any trouble. It is capable of evacuating as much as 100 cu. m. (130 cu. yd.) of alluvial matter a day.

When at the beginning and end of the summer the amount of water in the torrent diminishes, the alluvial matter it carries decreases in quantity very rapidly, and it is possible, by partially closing the sluice V_4, to cut down the water passing to the flushing channel till it just suffices to carry off the small amount of alluvial matter that is deposited, while, if the very small quantity of water coming down the stream demands it, the sluice V_4 may be closed entirely. Experience has shown that if, after running for some time in this way with this sluice closed it is opened again, the small amount of sand and mud which will have been deposited on the sluice V_4 is closed will rest on the slats between them, so that in the event of the periodic opening of the sluice being forgotten the deposit does not pass into the orifices themselves or into the flushing channel. If the separator is emptied these deposits can be washed away just as easily as those formed in a decanting basin with a flat bottom. From its first installation the orifices of the Gornergrat separator have never been covered with deposit, and it has not been necessary to empty the basin to wash away settled matter. When an eliminator is arranged in connection with several decanting basins it would no doubt be possible to empty them in turn for a washout without any interruption in service, but in order to spare himself the trouble of emptying them and the operation of the many valves which would be necessary for this, the man in charge would willingly undertake to open the sluice V_4 from time to time to take advantage of the automatic flushing arrangement.

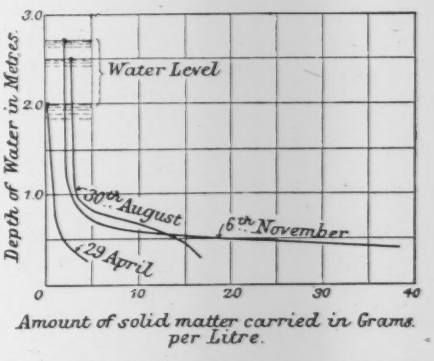


FIG. 9 DISTRIBUTION OF ALLUVIAL MATTER THROUGH THE CROSS-SECTION OF RIVER FLOW

This eliminator was put into service in June, 1921, and notwithstanding rapid variations in the amount of water in the stream and very great quantities of alluvial matter, has kept the plant going uninterruptedly ever since. It is also claimed that the operation of the turbines has been considerably improved and the wear has become very small. A number of other installations of a similar character have been made in various parts of the world, ranging as high as 20,000 hp. In principle the arrangements described are suitable for water consumptions up to 3500 cu. ft. per sec. Where the dimensions of the plant would make it cumbrous and costly, special arrangements may have to be adopted. In rivers and canals the density of the alluvial matter is much greater near

the bottom than at the surface. The concentration of the deeper strata applies particularly to the larger grains of material which have the most serious effect on the turbine. The results of sampling carried out to investigate this matter by M. R. Boissier on the River Arve at Geneva are plotted in Fig. 9. This characteristic of the larger particles of alluvial matter makes it possible in plants of medium or low fall to limit the treatment of water to the elimination of grains above a certain size and to employ an arrangement of automatic eliminator which, while it prevents the formation of deposits, makes it possible to treat very large quantities of water in an installation of relatively small size. A plant of this kind designed for Japan is described in the original article. In this a shelf situated some height above the bottom of the canal separates the water flowing near the bottom of the channel from that above it and turns the lower water into the automatic eliminator. The water passing through this latter, which is only a fraction of the total, rejoins the main stream leading to the power house by a certain passage, while the alluvial matter is carried past into the river (Henri Dufour, in *Engineering*, vol. 120, no. 3115, Sept. 11, 1925, pp. 313–316, *d*)

INTERNAL-COMBUSTION ENGINEERING (See also Motor-Car Engineering: The Knecht Turbo Atomizer)

Valve Gear in Engine of the Imperia Car

Data of a Belgian car having an engine with an unusual valve gear. The valves themselves consist of curved slides which are reciprocated by camshafts and rocking levers. On each side of the cylinder is a longitudinal recess of which the back is broached out to an arc concentric with the cylinder bore. The sides of the recesses are radial, so that the slide valves cannot drop out into the cylinder proper when the piston is removed. Two ports are cored through the cylinder block near the upper end of each recess, the ports on one side communicating with an external inlet manifold, and those opposite with the exhaust manifold.

In each recess there is a slide valve the back of which rubs against the broached surface, while the face is ground true with the bore of the cylinder. Ports in the valves coincide with the ports in the cylinder as the valves are reciprocated at half crankshaft speed, the opening and cut-off being timed more or less as in the case of a conventional poppet-valve gear.

The slide valves are made of cast iron and are approximately 40 mm. by 5 mm. in cross-section, with ports 30 mm. by 6 mm. At the lower end of each slide are lugs carrying a trunnion pin, engaged by a forked rocking lever. This lever is pivoted on a longitudinal shaft fixed in the crankcase, while a second fork in the lever embraces the crankshaft. To move the slide valve upward, a cam engages a roller on the rocker, but for the downward movement a second cam engages a flat surface on the fork. The controlled motion of the valve is not of a harmonic nature, as in the cases of the Knight and Burt sleeve valves, neither is it so irregular as that of the ordinary poppet valve. Its motion is such that the valve is already in rapid movement as the ports are opened, and is still moving sharply when they close. Thus the diagram of effective valve area shows to great advantage when compared with that of an ordinary cam-operated poppet valve, the steepness of the opening and closing curves of which is limited by considerations of inertia, stress, and noise. (*The Automobile Engineer*, vol. 15, no. 206, Sept., 1925, pp. 278–283, illus., *d*)

MACHINE PARTS

The Spiro Ball Bearing

Description of a ball bearing manufactured by the Spiro Ball Bearing Co., Ltd., Kettering, England, for which it is claimed that the load-carrying capacity of a roller bearing is obtained while the low coefficient of friction incidental to the use of balls is retained. In this bearing the balls are arranged in a helix, any load being taken by arranging a sufficient number of convolutions in the helix. The inner and outer races of the bearing are cylindrical and the balls are housed in a cage consisting of a light-alloy die casting.

It is claimed that with the helical arrangement no two balls run in the same path, so that wear over the bearing surface is equalized. It is also claimed that a self-lubricating property is attained. Each ball squeezes the grease into the path of the ball immediately following or preceding it, so that each ball is continuously supplied with grease by its neighbors. It is claimed also that this feature has the effect of retaining the grease in the bearing, so that it may be used with safety in such places as textile mills, boot factories, etc., where grease dropping from the lineshafting would spoil the goods. (*Mechanical World*, vol. 78, no. 2017, Aug. 28, 1925, p. 160, 4 figs., *d*)

MECHANICS

Torsional Vibrations in Reciprocating-Engine Shafts

In the calculation of the location of critical speeds of shafts from the constants of the machinery, the reciprocating parts of the engine are replaced by an "equivalent mass" which is assumed to contribute to the elastic vibrations of the shaft in exactly the same way as does the actual rather complicated system of crank, connecting rod, piston, and piston rod. The purpose of the author is to examine the correctness of this equivalence, and it is stated in the paper that the results obtained are applicable only to relatively low-speed machinery.

The author examined analytically the vibrations about steady motion of an elastic shaft having an engine at one end and a mass operating against a resistance at the other. The engine was assumed to be of the reciprocating type, and to fix ideas it was assumed to be of the six-cylinder variety of four-stroke single-acting Diesel engine.

The engine parts were assumed to be rigid. The inertia of the reciprocating parts of the engine was fully taken into account, but no account was taken of the internal resistance to vibration in the flexible shafting of resistance to vibration which must occur at the load end of the shaft.

The oscillations about steady rotation are found to be of two types, analogous to free and forced vibrations, respectively. The forced vibrations do not arise entirely from the pressure of the expanding gases on the pistons. They are also produced by incomplete balance of the reciprocating parts. Practically all the phenomena associated with dangerous critical speeds would appear if the fuel were cut off and the engine made to run against no resistance with the required speed.

The vibrations, analogous to free vibrations, appear to be of the form $e^{\sigma t}$ multiplied by periodic terms, where σ is a small contant and t is the time. Throughout the whole range of possible speeds of rotation at least one of the quantities σ appears to have a real part, existing and positive. This would point to the fact that, apart from frictional resistances, the vibrations everywhere tend to increase in amplitude with time. As this is contrary to experience, it is probably that the various kinds of frictional resistance that are possible have the effect of cutting out the real and positive part of σ, or, if not exactly canceling it, at any rate of leaving the real part negative. This would cause any such vibrations, if at any time excited, to be damped out.

The type of vibration produced is in general very complicated and cannot be definitely said to consist at any speed of a simple vibration of definite period. Two very important facts, however, appear:

(*a*) Suppose the applied force to have a term of the form $P \sin 3\omega t$, where ω is the mean angular velocity of the shaft and t is the time. This would correspond to three vibrations in the period of one rotation of the shaft. According to the usual theory, a critical speed would occur whenever the natural frequency of elastic vibration of the system was three per revolution of the shaft. It is shown, however, that the same term can produce a critical speed whenever the natural frequency of vibration of the shaft is, 3, 6, 9. . . per revolution of the shaft. This means that the existence of terms such as $\sin 6\omega t$, $\sin 9\omega t$ in the applied force is not essential, and their absence cannot be taken to imply absence of corresponding critical speeds.

(*b*) Under certain circumstances, depending upon the magnitudes of the parts of the system, in the vicinity of a critical speed corresponding to six vibrations of the shaft per revolution we may

find that the curve is not at all like the curve of sin $6\omega t$, but is almost exactly sin $3\omega t$. The critical speed is right and the vibrations large in amplitude, but the period of the curve of vibrations is twice that anticipated. This explains the phenomenon found in an actual case and automatically registered on a torsiograph. (G. R. Goldsbrough, D.Sc., Armstrong College, Newcastle-on-Tyne, in *Proceedings of the Royal Society*, Series A, vol. 109, no. A 749, Sept. 1, 1925, pp. 99–119, 1 fig., *m*)

MOTOR-CAR ENGINEERING (See also Internal-Combustion Engineering: Valve Gear in Engine of the Imperia Car)

The Knecht Turbo Atomizer

THE main part of the atomizer is a turbine wheel consisting of a part which looks like a tube with a number of longitudinal slots. This is held on a frame running in ball bearings. The aluminum casing of the atomizer has in addition to inlet and outlet ports for the fuel an air port controlled by an air-regulating throttle. The whole apparatus is supposed to be set between the engine and the carburetor. The carburetor controls as a rule do not need to be disturbed, while the air throttle of the atomizer may be set on the control board of the car.

Where the atomizer is installed the carburetor nozzles have to be readjusted, smaller nozzles being as a rule required. It is claimed that the presence of the vaporizer produces a certain precompression of the mixture, as well as a better mixing of the fuel and air. (*Der Motorwagen*, vol. 28, no. 23, Aug. 20, 1925, pp. 507–508, *d*)

POWER-PLANT ENGINEERING

Tests on a Cooling-Water Evaporator

A COOLING-WATER evaporator is employed to utilize the heat which the cooling water in a condenser collects from the condensation of the steam, the heat recovery being secured by means of evaporating part of the cooling water and using the distillate for boiler feeding. Without this heat-recovery device the heat in question would be completely lost. The principle of operation of the device lies in conveying the cooling water from a condenser to an evaporator connected with an auxiliary condenser in which a higher vacuum is maintained than in the respective main condenser. With this arrangement a part of the warm water in the evaporator is vaporized, the amount of water so vaporized being such as to reduce the rest of the water to the temperature corresponding to the vacuum obtaining in the evaporator.

The practical execution of this principle is primarily dependent on a suitable arrangement of the auxiliary condenser by means of which it becomes possible to maintain actually the higher vacuum necessary. Several designs have been proposed for this purpose in Germany. In the Josse-Gensecke device the auxiliary condenser is inserted on the cold-water side of the main condenser ahead of the latter and in the line of the cold-water flow. This arrangement is particularly suitable when fresh water is used in the main condenser (Fig. 10).

In the Balcke system (Fig. 11) a separate cooling-water circulation is maintained with a lower temperature level than the main cooling-water circulation. This arrangement is used mainly when the main condenser is operating from a cooling tower. To achieve it the auxiliary condenser may be fed with fresh water, its consumption being very small as compared with that of the main condenser; or else special precooling, for example, by a fan, may be used on the auxiliary condenser water.

Another arrangement which may be used is to set the evaporator between two condensers operating with different vacuums; for example, the main general condenser and the turbine condenser, or condenser operating with cooling-tower-cooled water and condenser operating with fresh water. In large power plants facilities for doing this are quite often available. A further advantage to be found in this arrangement is that it makes unnecessary a separate auxiliary condenser and thus reduces the cost of the installation.

The tests described in the article have been carried out on an installation arranged essentially in accordance with Fig. 11, and the original publication gives a description of the experimental installation and data of tests in the form of tables. From these it would appear that the thermodynamic efficiency of the evaporator is determined by the ratio of the actual temperature difference in the evaporator to the temperature difference theoretically possible therein. By "temperature difference" is here meant the difference in temperature between the water at the inlet to and the outlet from the evaporator. From data collected in the course of these tests it would appear that the best results from every point of view are obtained when this temperature difference is of the order of 10 deg. cent. (18 deg. fahr.). Under these conditions the minimum of cooling water is required in the auxiliary condenser and the minimum practical amount of salts contained in the water is carried

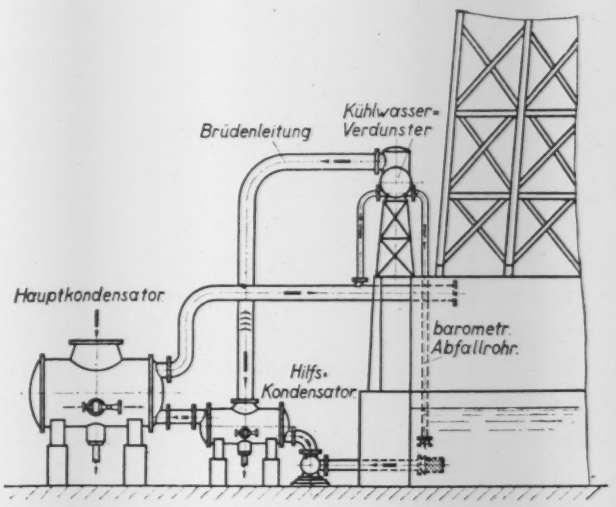


FIG. 10 JOSSE-GENSECKE CONDENSER-COOLING-WATER EVAPORATOR WITH A SINGLE WATER CIRCULATION

(*Hauptkondensator*, main condenser; *Hilfs-Kondensator*, auxiliary condenser; *Brüdenleitung*, vapor piping; *Kuhlwasserverdunster*, cooling-water evaporator; *Barometer-Abfallrohr*, barometric discharge pipe.)

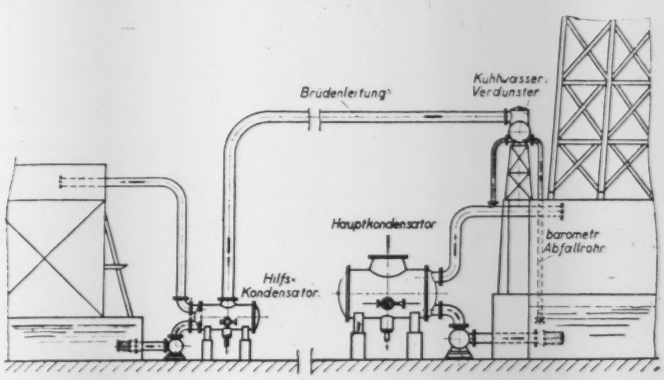


FIG. 11 BALCKE CONDENSER-COOLING-WATER EVAPORATOR WITH TWO SEPARATE WATER-CIRCULATION CYCLES

(For translation of the German words, see caption of Fig. 10.)

off with the distillate. (Dr. of Engrg. L. Heuser and Engr. H. Kanzler, Bochum, in *Die Wärme*, vol. 48, no. 28, July 10, 1925, pp. 355–358, 7 figs., *e*)

Pumps and Injectors for Feeding Steam Boilers

THE author carried out some tests on boiler feeding by means of pumps and injectors and presents data of these tests. The conclusions to which he comes are that—

1 If there is no use for the exhaust steam from the pumps and the flue gases are not utilized in a preheater, injector feeding is thermodynamically far superior to pump feeding. Considering, however, the small amounts of heat utilized in either case and the high rate of steam consumption of both means of feeding, it is a question whether the superiority of injector feeding will be maintained in spite of the possibility of steam losses through frequent cutting in and out of the injector.

2 Where the exhaust steam from the pumps finds a useful

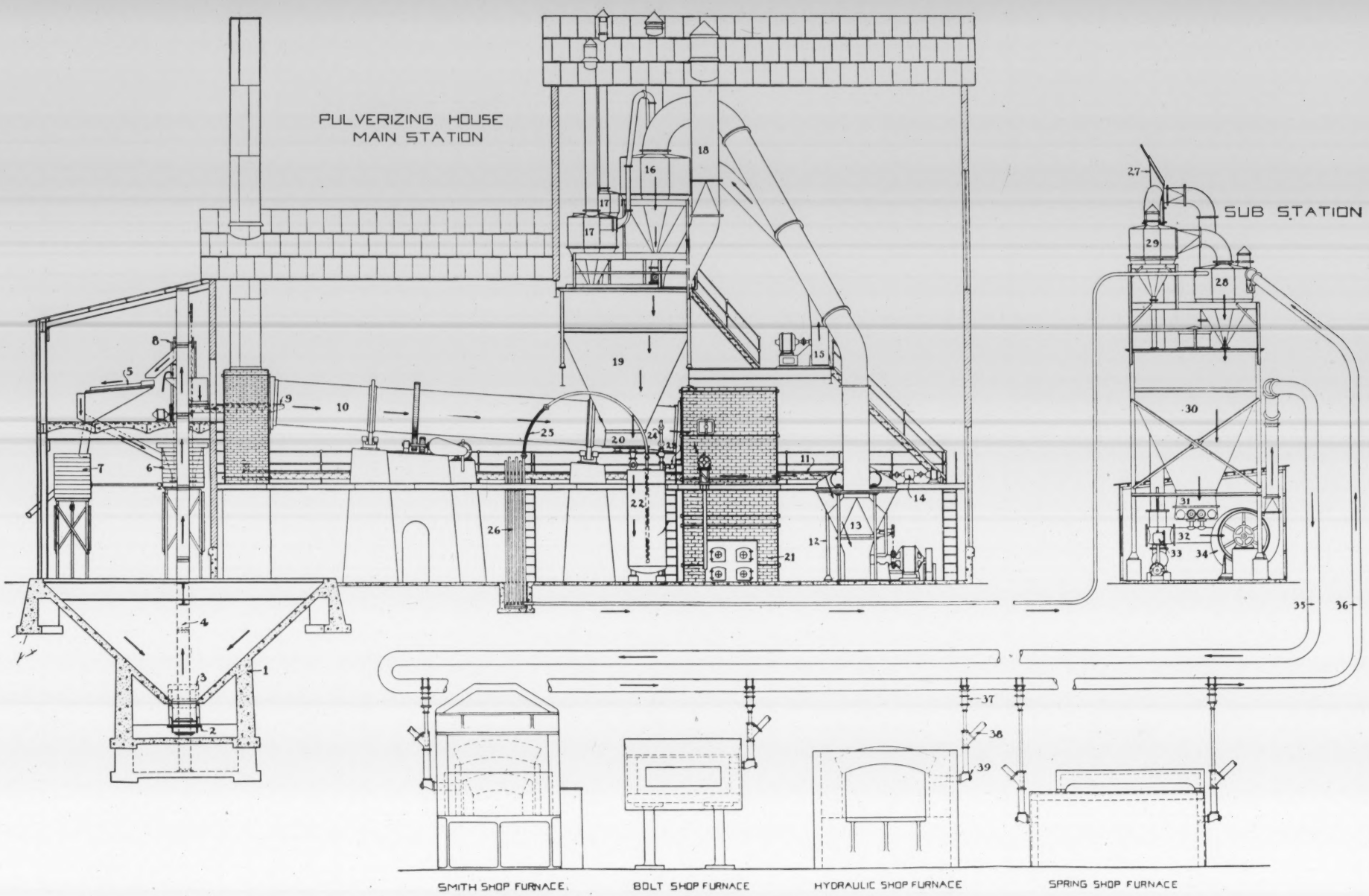


FIG. 12 POWDERED-COAL INSTALLATION AT THE BALDWIN LOCOMOTIVE WORKS

1 Track hopper
2 Apron feeder, coal from track hopper to crusher
3 Four-roll coal crusher
4 Bucket elevator
5 Coal-washing table
6 Washed-coal dewatering tank
7 Sludge or waste tank
8 Dewatering conveyor to conveyor screw entering drier
9 Screw conveyor to rotary drier
10 Rotary drier
11 Screw conveyor from drier to 5-ton storage bin
12 Dried coal, 5-ton storage bin
13 Coal pulverizer and separator
14 Feed-screw speed reducer
15 Exhauster, powdered coal from separator to collector
16 Powdered-coal collector
17 Auxiliary collectors
18 Safety vent for 25-ton bin
19 25-ton pulverized-coal storage bin
20 Feeder box
21 Furnace for drying crushed coal in drier
22 Ejector tank
23 Compressed-air line
24 Air relief
25 Flexible-hose connection
26 Distributing lines to substations
27 Safety vent
28 Powdered-coal collector
29 Auxiliary collector
30 Powdered-coal storage bin
31 Feeder box
32 Coal-feed pipe to blower
33 Automatic regulator
34 High-pressure distributing blower
35 Distributing line to furnaces
36 Return line to collector and storage bin
37 Butterfly valve
38 Secondary air line
39 Powdered-coal burner.

application but the flue gases are not utilized in a preheater, the theoretical efficiency of both methods of feeding is the same. For practical reasons explained above, pump feeding is, however, superior.

3 Where flue gases are utilized in a preheater, the thermodynamic economy of pump feeding is so superior that this device is worth using, even where no use can be found for the exhaust steam from the pumps, and this notwithstanding the high steam consumption of the pump and the low efficiency of the preheater. (Karl d'Huart in *Feuerungstechnik*, vol. 13, no. 23, Sept. 1, 1925, pp. 277–278, *ep*)

Powdered-Coal Installation at the Baldwin Locomotive Works

UP TO THE close of the World War oil was used for furnaces at the Baldwin Locomotive Works. However, there were times when not only prices were uncertain but delivery as well, which led the plant to an investigation of possibilities of other fuels, and to the final decision to use powdered coal.

The present powdered-coal installation comprises a main station with seven substations, handling the full requirements of 160 furnaces, four Babcock & Wilcox boilers of 600 hp. each, and three pile-heating furnaces. In the design of the main station wherein is housed the equipment for crushing, drying, pulverizing, and ejecting the coal to the substations, every precaution was taken to eliminate any unnecessary ledges or projections on which coal dust could accumulate. The electric-light installation is of the vapor-proof marine type; motors for all machines and feed screws throughout the building are operated from the floor by a remote-control system. All equipment as well as the side walls of the building have been painted battleship gray in order to make discernible any accumulation of coal dust, which is removed on account of fire hazard. Fig. 12 shows the course of coal from the track hopper to the furnace. In many respects the equipment is of the conventional type. The crusher is of the four-roll bypass spring-relief type, the upper rolls having teeth and breaking the lumps to a uniform size of 2 in. The lower rolls are corrugated and reduce the coal to such size that about 90 per cent will be $^1/_2$ in. and under. The pulverizers are of the Bonnot type. Powdered coal is carried by a stream of air through the distributing chamber and up through the flues to adjustable deflectors which permit the lighter particles to go on and send the heavier ones back into the pulverizer. The stream of air and fine coal then passes through a collector which separates the air from the coal and sends the latter into the 25-ton storage bins. The coal considered suitable in this instance has a fineness of 85 per cent passing through a 200-mesh screen, with the remainder through a 100-mesh. From the bottom of the storage bin the coal passes through a valve into the ejector tank and is then ready to be delivered to the substations through a 3-in. pipe under air pressure. There are seven of these pipes, one for each substation, and coal can be fed to two substations simultaneously by using the two ejector tanks. Flexible hose is attached to each discharge pipe and connections can be made from either of the ejector tanks to any one of the seven pipes leading to the substations. The air pressure required to force the coal through the pipes to the various substations varies according to the distance to be travelled, ranging from 20 to 50 lb.

The type of burner and method of ejecting the air into the branch line is shown in Fig. 13. The low-pressure air is brought by an independent branch line and meets the coal which is brought by the distributing line, and a sufficient amount of air must be passing through the branch line to carry the required amount of coal to the point where it meets the secondary air. Combustion within the furnace is controlled by the secondary air.

When the plant is in operation and the furnaces are shut down between shifts or for other reasons, the substation blower is run for a short time to clear all pipes of powdered coal and return it to the storage bin. This return system eliminates all danger from coal settling in the pipes and causing a "puff" at the furnaces or boilers when the operator lights his fire. Another feature is that all substations are outside the buildings wherein the coal is used. With the coal in circulation returned to the storage bin, there is no fire hazard within the shops.

The furnaces are so arranged that either powdered coal or oil can be used at very short notice. The boilers were previously equipped with Murphy stokers. When these stokers were removed it was found that the boilers needed various alterations to obtain sufficient combustion space for burning powdered coal. At present boiler ratings of 125 to 175 per cent are not unusual and peak loads can be met readily. However, frequent cleaning of the bottom of the combustion chamber and blowing down of the flues is necessary if the greatest efficiency is to be maintained.

The Deister-Overstrom table is used for eliminating ash and foreign matter entering the coal. Crushed coal with sufficient water is delivered to the table through the feed box. Water distributing passages are provided along one side of the table. In eliminating or washing the refuse from the coal, advantage is taken of the difference in their specific gravity. Crushed coal passes over the riffles in the table, the heavier particles, or refuse, slide along the bottom, and the clean coal, being lighter in weight, is washed over the top. The table, when in use, is placed in an inclined position, the degree of inclination being lessened or increased in accordance with the amount of coal required, its capacity ranging from seven to ten tons per hour. By mechanism erected underneath, the table is given a reciprocating motion of 275 strokes per minute, with a stroke length of $^3/_4$ in. This motion, which causes a stratification of the coal and foreign matter, also drives the foreign matter from under the clean coal and over to the refuse edge of the table, from which it drops into a refuse storage tank. The clean coal passing over the edge of the table is carried by screw conveyors to the dewatering tanks and thence to the drier.

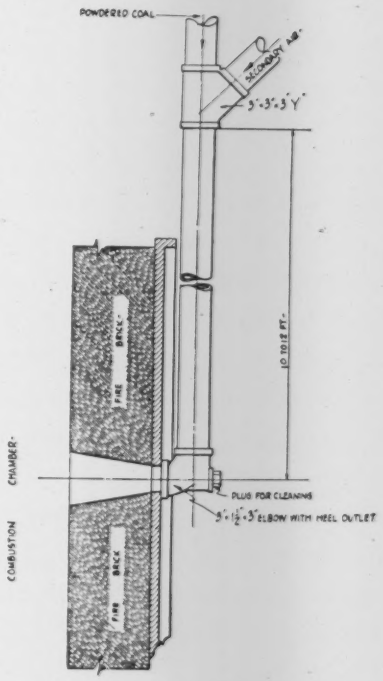


FIG. 13 POWDERED-COAL BURNER AT THE BALDWIN LOCOMOTIVE WORKS

Not only does the washing table remove wood, tramp iron, etc., but the following large percentages of ash and sulphur are eliminated as shown by reports:

	Ash	Sulphur
Raw coal	9.30	2.10
Washed coal	6.74	1.63

The following statement is made as to costs: If the installation is run at or near its rated capacity, i.e., with a high load factor, the cost per B.t.u. in the furnace with powdered coal is lower than with fuel oil at the fuel prices ordinarily prevailing. For example taking the cost of a good grade of gas coal, including pulverizing, to be $5.50 per ton, and that of fuel oil, 5 cents per gallon, the purchase value of one cent expressed in B.t.u. would be as follows:

Powdered coal	50,018 B.t.u.
Fuel oil	27,600 B.t.u.

However, it must be remembered that the true measure of boiler or furnace efficiency is one of heating cost and not of fuel cost. Powdered coal was not decided upon merely because it showed a lower B.t.u. cost, but because the production cost, under the company's shop conditions, was found to be lower than that with any other fuel.

Cleanliness, ease of operation, no loss of coal through the grates, no loss of heat to remove the moisture from the coal in the combustion chamber, close control of fluctuating loads, ability to use a cheaper quality of coal, lower expenditures for labor—all these are factors which recommend the use of powdered coal for fuel. (Carl C. Bailey in *Baldwin Locomotives*, vol. 4, no. 2, Oct., 1925, pp. 37–46, illustrated, *dA*)

RAILROAD ENGINEERING

Electric Locomotives on the Ford Road

THE new electric locomotive for the Detroit, Toledo, and Ironton Railway, which involves certain novel features of mechanical design, has been completed and placed ready for tests. (Its electric system was described in MECHANICAL ENGINEERING, vol. 47, no. 1, Jan., 1925).

The locomotive is primarily a freight engine of 0–8–8+8–8–0 wheel arrangement. It consists of two motive-power units semi-permanently coupled together, each one carrying its own full equipment. Each motive-power unit consists of two articulated eight-wheel trucks connected by a universal hinge of the ball type. The leading truck carries chiefly the auxiliary equipment, while the trailing truck carries the 750-r.p.m. motor-generator set. Each power unit has eight traction motors of the direct-current 600-volt series axle-hung type of 225 hp. each. Each motor is geared to the driving axle by a spur-gear transmission.

The journals were given generous proportions to minimize deflection of the axle and make a quiet-running gear. Both pinions are flexibly connected to the motor by a quill drive of simple and efficient design. The hollow motor shaft carries the armature and rests in two bearings of standard design. Inserted and keyed to it internally at half-length is a flexible propeller shaft which has fastened to each end a driving pinion. These pinions again have shanks fitted loosely in the hollow armature shaft. Such arrangement permits the pinions to rotate relatively to the hollow shaft as well as to each other just enough to insure a good mesh with both gears, and absorb any vibrations in the drive. This mechanism was submitted to quite a severe run on the test floor—equivalent to about two years' road service—and proved very satisfactory.

Each motor unit is suspended at three points and guided at an additional fourth one in the following way: The tractive force of each motor is transferred to the locomotive frame by a universal ball-and-socket joint while the weight of the upper locomotive structure rests on two equal beams located directly above the axle. Two brackets reaching out from the motor housing serve as supports for the equal beams. In order to keep the motor units lined up on the track and prevent any lateral motion, a point opposite the universal suspension joint with respect to the axle is guided vertically in a frame extension. Only six motors for each power unit, however, are assembled this way, while the leading and trailing axles have spring-buffered guides to allow a maximum of 2 in. lateral motion each way in curves. The equal beams mentioned above rest on rocker pins and are flexibly suspended on a series of leaf springs, the whole system forming a spring rigging with five fixed points for each truck.

The cab is constructed in the usual way but made of heavier stock than necessary. The outside walls are $^1/_4$-in. plates. Partitions and inside walls for the double-walled engineman's compartment are $^3/_{16}$ in. thick. The two cab halves are joined by a flexible canvas diaphragm consisting of six metal-reinforced folds suspended by two universal ball joints from the cab roof. The center portion of each half-cab can be lifted away, while the engineman's compartment and the section nearest to the articulation remain in place. This feature facilitates work in case of heavy overhauling.

A radical departure from common practice was made in the air brake. Large air cylinders, complicated linkages, and side brake shoes were replaced by individual brakes for each wheel except for those of the leading and trailing axles. The object of this change was to do away with all cumbersome obstructions which are involved with the use of many links, levers, and so on. The new design provides for a small air cylinder on top of the wheel supported on a traction-motor-housing extension. With compressed air admitted, cylinder and piston travel in opposite directions, forcing a single brake shoe down upon the wheel tire through multiplying levers. A flat leaf spring releases the brakes when air is exhausted. The forces exerted by this mechanism will not affect the cab springs but will merely relieve the axle journals of a certain amount of weight, or, in other words, a weight transfer will take place from the journal to the top of the tire. Most of the parts are interlocking and may be taken apart with but few tools and in a few operations. All other parts of the air-brake equipment are of standard design. (*Railway Age*, vol. 79, no. 9, Aug. 29, 1925, pp. 389–392, 5 figs., *d*)

New Rail Section of the Reading Company

THE Reading Co. is trying out a new 130-lb. rail of the section shown in Fig. 14. The flange and web are identical with the 130-lb. A.R.E.A. section. An area of 0.135 sq. in. is taken from each of the lower corners of the head and added to the top, where $^3/_{32}$ in. greater depth for wheel abrasion is thus supplied. It is also believed the new section is better than the A.R.E.A. for rolling. It does not require the fishing angles under the head; it permits a much better working of the steel in the head which, in turn, gives better texture and should give longer life. The distribution of metal as between head, web, and flange is exactly the same as in the A.R.E.A. section. The redistribution, however, gives a higher moment of inertia.

Two factors stand out preëminently in considering the modification of the A.S R.E. section. The first is the facility with which the superfluous metal may be removed from the lower corners of the head and added to the top of the rail. Small as it is, this amount of metal is sufficient to extend the life of the rail approximately

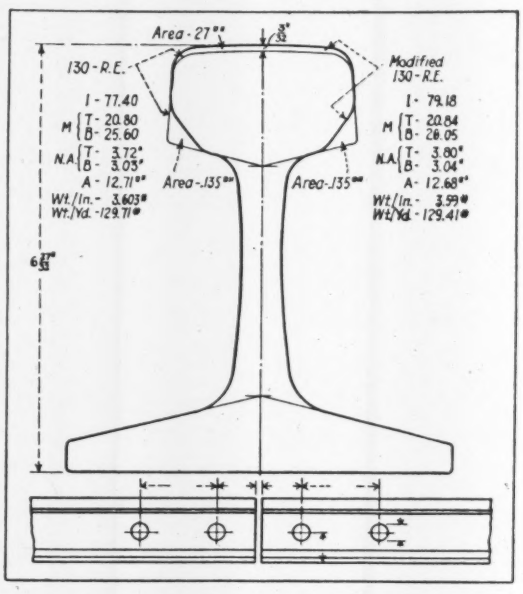


FIG. 14 FORM AND PROPERTIES OF THE NEW RAIL SECTION OF THE READING COMPANY

20 per cent. It has the advantage that it does not disturb the head, web, and base ratios of the A.R.E.A. series of rails, which rightly are becoming so popular. The added life is proven by careful readings which have been made of the area of metal abraded from the head of each standard 130-lb. R.E. rail on a 9-deg. spiraled curve which has 4 in. superelevation. When this standard rail was taken out of the track at the end of two and a half years it was found that the area of the $^3/_{32}$ in. of additional metal on the head (0.27 sq. in.) would be equal to at least one-fifth of the total area which had been abraded.

The second factor is that the constantly undiminishable splice bearing areas on the head fillets on the rail, when the head-free joint is used, supply much more effective and permanent bearings for the splice bars than have been secured heretofore by the prevailing type of flat fishing bearings on the under side of the rail head.

The first makes for material economy in rail expenditures and the second for a substantial economy in maintenance and preservation of the rail ends. (*Railway Review*, vol. 77, no. 13, Sept. 26, 1925, pp. 469–470, 3 figs., *d*)

SPECIAL MACHINERY

The Carpenter Centrifuge for Coal Drying

THE demand for an economical method of drying of coal has been accentuated in recent years by the rapid development of coal-washing processes. In this connection the Carpenter centrifuge recently installed at the Nunnery Colliery Company's pit in Sheffield is of interest.

This centrifuge comprises a stepped truncated cone driven by a vertical shaft and enclosed in a circular casing. The wet coal is continuously delivered into a stationary hopper and falls through it on to a horizontal revolving distributing disk fixed to the vertical shaft, which latter can be so adjusted that the orifice between the cone and the edge of the disk insures an even distribution with varying qualities of coal. The coal is thrown from the distributing disk by the centrifugal force against the top screen. A portion of water passes through the screen. The partially dried coal travels down on the inside of the screen by centrifugal force until it reaches the first cascade. Here it is broken up by serrated teeth and redistributed evenly to a second screen. More water is extracted and the coal slides down the screen to the next cascade, and so on.

The screens are made of 1/8-in. steel plates with 1/8-in. holes 3/16 in. between centers. The number of cascades is varied to suit the particular material. It is claimed that a thin layer of coal continuously moving is always retained on the screen surface, and this, although not thick enough to prevent the passage of water, acts as filter and stops the egress of fine coal. It also protects the screen plates from wear.

At the Nunnery pit the machine readily handles about forty tons per hour, reducing the moisture content of the dried coal uniformly to between 6 and 7 per cent. The power consumption is under 40 hp. The labor required is very small as the machine runs for hours without attention. The machine has been on the market for a considerable time and is manufactured in England. (*The Iron and Coal Trades Review*, vol. 111, no. 3001, Sept. 4, 1925, pp. 359-360, 2 figs., *d*)

TESTING AND MEASUREMENTS

Hardness Test Research

REPORT on the effects of adhesion between the indenting tool and the material in ball and cone indentation hardness tests of the Hardness Test Research Committee of the Institution of Mechanical Engineers (British). The report contains the results of experiments which have been made with the object of supplementing existing knowledge on the results given by static ball and cone indentation hardness tests when adhesion between the indenting tool and the material is considered. The investigation involved first miscellaneous tests showing the existence and effect on the adhesion, and next ball and cone tests in which a correction for the adhesive forces is applied to the results.

It was found that adhesion exists in the normal ball indentation test and that its effect is greater in the deeper impressions. It also depends to a limited extent on the lubrication conditions existing in the test. The adhesive forces were found to be proportional to the applied load, and it is believed a constant ratio of the tangential adhesive forces to the normal forces in an indentation test might be obtained; a similar ratio, in fact, to the coefficient of friction. By employing certain assumptions, an expression has been obtained by means of which, under certain conditions, the main resistance to indentation corrected for the adhesion can be calculated from the diameter of the indentation measured after test.

The formula obtained was tested and found to work out well. From this it would appear that adhesion is the direct cause of the high values of the uncorrected contact pressures given by the smaller-angle cones.

As regards cone tests, it was found that for any given cone and material, and constant initial surface conditions in regard to lubricant, the value of the contact pressure is independent of the load. The value of the corrected constant pressure is found to be reasonably constant for a particular material and independent of the angle of the cone or the applied load. This general result, however, does not apply to manganese steel.

In the ball test the corrected contact pressures (except on manganese steel) have an approximately constant value for a particular material over a certain range of load, which range is dependent on the size of the ball and the material under test. Further the constant value over this range of load is in fair agreement with the corrected contact pressure given by the cone tests on the same material.

The corrected contact pressures obtained in several different ways and expressed as a pressure in kilograms per square millimeter agree fairly well with the standard Brinell hardness numbers for all the materials except manganese steel. The reason for this is fairly obvious; the use of the spherical area in the standard test makes a difference in the value of the contact pressure which is of the same order as the adhesion correction, and hence the standard Brinell numbers are approximately the same as the corrected ball and cone contact pressures.

The existence of an approximately constant value of the corrected contact pressure over a particular range of load is of considerable interest. It will be noted that the range of load varies for different materials commencing at low loads on the rolled copper and work-hardened steel, and at about 3000- to 4000-kg. load on the remaining materials (except manganese steel) for the 10-mm. ball tests. For the 5-mm. ball tests the loads are proportionately lower. It is usually considered that the increase in ball hardness numbers at the higher loads is due to increased resistance to penetration following on the cold working of the material during a test. The present results show this only occurs up to a certain value of the load, which is dependent on the material; beyond that load further cold working is not apparent, and the resistance remains approximately constant. There is a tendency for the corrected contact pressures to fall at very high loads in the ball tests, but it can hardly be expected that the adhesion correction is valid when a portion of the surface of the ball is nearly parallel to the line of action of the applied load; a condition which occurs when the ball is deeply embedded in the material. In the case of manganese steel, no value of μ could be found which gave agreement of the various cone-test results. No direct evidence was obtained on which to base a value of μ, but it was considered that $\mu = 0.3$ would show the general trend of the results. It is inferred from the corrected results on this material that continued cold working has occurred at all loads, and it is very doubtful if the maximum value of the corrected contact pressure can be obtained in a ball test. The cone-test results, showing increased contact pressures with decrease in the vertical angle of the cone, also support this inference, which is consistent with the known properties of the material.

In general, from the results of the present investigation, it appears that the resistance to indentation of a material by a ball or cone, when corrected for adhesion between the material and the ball or cone, is reasonably independent of the applied load and the indenting tool, provided the vertical angle of the indenting cone is not greater than a certain limiting value, and the ratio of the ball diameter to the impression diameter is not greater than a corresponding limiting value. The limiting values specified above are dependent on the test material, and are apparently related in some way to the degree of work hardening which has been given or can be given to the material. It seems likely that the ratio should be independent of the ball diameter for a given material. For the materials considered, excepting manganese steel, the 120-deg. cone appears to be less than the maximum limiting cone angle.

Dr. Unwin has given a definition of hardness as "the intensity of pressure at which the material yields or flows in the conditions of the indentation test," but probably because of the differences obtained with various indenting tools and variations with load in ball tests, his definition does not appear to have been generally accepted. The corrected contact pressures given in the results represent a more rational value of the intensity of pressure than the usual ball-and-cone hardness numbers, and it has been shown that the value, within certain limits, is independent of the method of test. The hardness, expressed in this way, can therefore be regarded with greater certainty than has hitherto been the case, as a definite property of a material. During the last few years a large amount of experimental work on indentation hardness has been carried out, and attempts have been made to give a real meaning to the term "hardness." It is considered that the present results substantiate Dr. Unwin's definition of indentation hardness as a definite property of a material when the time effect during a test can be neglected. In static indentation tests on the harder materials the time effect is small when the load is applied for 30 sec., but for soft materials, and for all materials under impact conditions, presumably the rate of yield, or flow, and the intensity of pressure would not be independent. It seems probable that allowance for adhesion

between the indenting tool and the material in impact indentation tests would yield interesting and possibly valuable results. Whether the distribution of the corrected contact pressure over the indentation can be determined and the result correlated with the other mechanical properties of a material remains to be shown, but in the opinion of the author it is not at all impossible.

In the case of the 90-deg. cone test the initial conditions of the cone and the material are of importance when accuracy is desired, and unless care is taken to maintain constant initial conditions variations in the results on a uniform material may be obtained. Direct conversion of cone test result to ball test result requires a knowledge of the value of μ, but it appears that a value of $\mu = 0.3$ might reasonable be assumed over the Brinell range of 70 to 300 without causing any very serious errors. The following rule would then give agreement of ball and 90-deg. cone test results.

$$\text{Brinell hardness number} = \frac{\text{Cone hardness}\left(\frac{P}{\pi r^2}\right)}{1 + \mu \cot \theta} = \frac{\text{Cone hardness}}{1.3}$$

(G. A. Hankins, National Physical Laboratory, Teddington, England, in the *Proceedings of the Institution of Mechanical Engineers*, no. 3, 1925, pp. 611–645, 11 figs., *et.A*)

The Rotoscope

Description of an apparatus working on the principle of the stroboscope, but differing from the ordinary stroboscope in the use of a shutter, as a result of which the area of vision at full aperture is but slightly reduced and a very small device weighing less than 2 oz. gives the same service for both eyes as two slotted plates 2 ft. 6 in. in diameter would give for one eye. The main well-known objections to the stroboscope are therefore eliminated and a very portable form of instrument is obtained which can be held in one hand and applied to the study of rhythmically moving or rotating objects having speeds ranging from the lowest to those of the order of 20,000 r.p.m., and possibly higher.

The shutter, invented by A. J. Ashdown, technical manager of the Westinghouse Morse Chain Co., Ltd., Letchworth, Herts., consists of an ebonite or other cylinder barely an inch in diameter and a little over 4 in. in length. Two slots $^3/_8$ in. wide and $1^1/_4$ in. long pierce the cylinder. The peripheral movement of the shutter (which amounts to $^3/_8$ in., corresponding to an angular rotation of about 43 deg.) causes the aperture for vision through the slots to open and close. (Fig. 15.) Thin plates are, however, inserted horizontally in the slots, with the result that while the area of vision at full aperture is but slightly reduced, the angle of rotation required to open and close the aperture is very much lessened.

Two glimpses through the aperture are obtained per revolution of the shutter, so that to obtain 12,000 glimpses per minute, the shutter speed will be 6000 r.p.m. The shutter is driven by a clockwork motor the speed of which can be regulated by means of a ball governor controlled from a knurled head outside the casing. In conjunction with the controller there is a dial which indicates the speed of the motor spindle at any setting of the governor. This speed ranges normally from 50 r.p.m. to 100 r.p.m., but the range can be extended to 30/125 r.p.m. From the motor spindle the drive is transmitted to the shutter through a five-speed gear box operated by a simple draw-out spindle. The gear ratios provided are 10, 20, 40, 80, and 160 to 1. It is thus possible readily to double the speed of the shutter. The range of speed of the shutter thus extends from 500 r.p.m. continuously and progressively to 16,000 r.p.m., or, using the extended motor speeds, from 300 to 20,000 r.p.m., with a considerable overlap between each gear change. At 20,000 r.p.m. the glimpse frequency is 40,000 per min. and the glimpse period one thirty-three-thousandth part of a second.

As a development of the original form of shutter Mr. Ashdown has devised one in which the left and the right eye slots can be set either collinear, as in the original model, or be displaced through 90 deg. relatively to one another. In the second setting the shutter does not, of course, give simultaneous binocular vision, but alternate vision with the left and the right eye. The impression left upon the sight is, however, indistinguishable from binocular vision. If the speed of the shutter is not changed the 90 deg. setting does not affect the duration of the glimpse, but it doubles the frequency. As another development an electrically driven model is being devised. In this model the glimpse frequency will be 180,000 per min. and the glimpse period about four millionths of a second.

The instrument has been applied to various purposes, for example, the study of the interaction of engaging gear-wheel teeth and the slipping of belts in high-speed pulley drives. Also the vibration of high-speed shafts and the cause of a disparity in the output of sewing machines, all of one pattern, and supposed to be driven all in the same way and at the same speed.

The motion of a dragon fly's wings has also been investigated and it has been found that the wings, contrary to hitherto-made assumptions, are not flapped up and down but are caused to execute a definite conical movement. (*The Engineer*, vol. 140, no. 3637, Sept. 11, 1925, pp. 272–273, 6 figs., *g*)

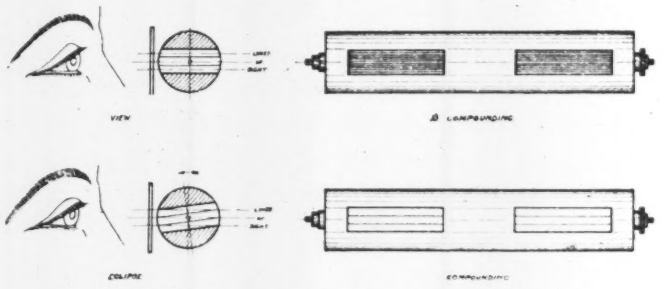


Fig. 15 Shutter of Ashdown Rotoscope

The Effect of Temperature on the Behavior of Metals and Alloys in the Notched-Bar Impact Test

Data of tests made on non-ferrous materials as part of an investigation on the behavior of a variety of materials both ferrous and non-ferrous. The tests were made in a 30-kg-m. Charpy machine. The British Engineering Standards Association standard form of test piece was used throughout, namely, 10 × 10 × 60 mm. (0.4 × 0.4 × 2.4 in.) with a 45-deg. V-notch, root radius 0.25 mm. (0.1 in.) and 2 mm. (0.08 in.) deep.

The following conclusions are drawn: Frequently a change in phase is accompanied by a change in temperature coefficient, resulting in an inflection or, in some cases, a maximum or minimum in the impact figure-temperature curve. A phase change is not always so indicated, and the inflection is usually not sufficiently sharp to be of use in fixing the temperature of the phase change other than approximately. Moreover an inflection does not necessarily imply a phase change. In alloys which retain considerable strength at high temperatures there is a marked rise in impact figure at a temperature at which recrystallization occurs instantaneously as deformation proceeds. On account of the much greater speed of deformation this temperature is much higher than the corresponding self-annealing or "recuperative" temperature in static tests.

The results suggest that a high notched-bar impact figure is an indication of good rolling properties. Maxima in the impact figure-temperature curves were found at the hot-rolling temperatures of zinc, lead-free 70:30 brass, 60:40 brass, aluminum bronze, and duralumin. If there is the connection indicated between impact figure and behavior on rolling, it follows that for the alloys named there is also an intermediate range of temperature within which their behavior on rolling should be worse than it is either at higher or lower temperatures.

Some comparative tests carried out from time to time on non-ferrous metals and alloys in the Charpy and the Izod machines lead to the conclusion that, if the test piece is broken, the energy absorbed, if not the same, is sufficiently close to enable relative results to be obtained equally well in the Izod and Charpy machines. If, however, the test piece is unbroken, there is often no agreement. In the Izod machine the standard test piece is bent through a maximum angle of about 60 deg., in the Charpy machine through an angle of about 130 deg. The relative absorption of energy in the range 0 to 60 deg. and 60 to 130 deg. may be so different in different material that they are placed in a different order in a series by the

two machines. Thus, for certain purposes the Izod machine is satisfactory, but for accurate measurements of energy absorbed it may be misleading, and a machine which gives a much greater angle of bend, such as the Charpy, is to be preferred. (R. H. Greaves, D.Sc., and J. A. Jones, M.Sc., in advance copy of a paper presented to the Institute of Metals in Glasgow, Sept., 1925, 17 pp., 13 figs., *e*)

Testing Dynamite

THIS work requires the utmost care, for a small miscalculation of weight or mixture can upset the entire operation. After the experimental dynamite is manufactured, it is placed in regulation cartridges, either by hand or by a one-stick Hall packing machine, which is operated identically as are the large machines employed for regular production work. The finished cartridges are finally packed in standard wooden cases and are then ready to embark upon their testing careers.

If the dynamite ordered is to be of a gelatin type, it is made up in the experimental gelatin mix house. Operated on a semi-plant scale, similar to the dynamite mix house, this house is outfitted with a miniature gelatin mixer, an exact reproduction on a small scale, of the regulation plant mixer. A gelatin packer is also part of the equipment.

There is no ordered sequence in which experimental dynamite is tested; the tests and their order are determined entirely by the chemists. Often, however, samples are taken and tested first by a lady in charge of the high explosives analytical laboratory.

In this building is also the Bichel gage for measuring and retaining for analysis the gases given off by dynamite upon explosion. The gage is in reality a sealed chamber, in which samples of dynamite are placed and shot. Cordeau fuse is also subjected to a separate test in this laboratory. The Mettagang recorder, a delicate piece of apparatus, records from the shooting grounds about half a mile distant the true velocity of the several grades of cordeau.

One of the most rigid and most recently developed tests that experimental dynamite is put to is the ballistic mortar test, conducted in a specially constructed building at the station. Small charges are accurately weighed in a laboratory balance and wrapped in small tin-foil cartridges. An electric blasting cap, with wires attached, is placed in one end of the cartridge, and it is ready for the mortar test. The finished cartridge is put inside an iron slug and the slug placed in the mortar, which latter is the base of a pendulum of aluminum frame construction, suspended on a knife edge. Wires running from a blasting machine in the adjoining room and down the pendulum are hooked up with the blasting-cap wires and the testers repair to the firing room. The safety door is closed. It is of a sliding type and must be closed to uncover the blasting machine, built in the wall. For reasons of safety, the tester who has hooked up the charge must fire the shot, and the slug is fired from the mortar into a pile of sacks in a protected tunnel. As the pendulum swings back, the strength of the explosive is determined by a pencil attached to the pendulum, which automatically draws a line on a graph sheet.

Tests for sensitiveness and rate of detonation are conducted on the shooting grounds, situated in a wooded portion of the property. The chief tester cuts the dynamite sticks into 4-in. lengths and, after wrapping them in cartridges, inserts blasting caps into them and gives them to his assistant who takes them (usually three at a time) to the shooting grounds, about 50 ft. from the dynamite shooting house, hooks up the cartridges, returns to the shooting house, and fires the charges in succession by an electric blasting machine, first ringing a gong to warn any persons in the vicinity of the coming explosions. The lead-plate tests for rates of detonation are conducted by these men. (*The Hercules Mixer*, vol. 7, no. 19, Oct., 1925, pp. 213–214, *e*)

THERMODYNAMICS

The Kinetic Theory of Evaporation

THE PHENOMENON of evaporation from liquids has long been explained by the kinetic theory on the assumption that various molecules are moving with different velocities. At a given temperature the faster-moving molecules tend to escape, leaving the remainder which have on the average slower motions. If they all moved with the same speed the liquid would have had no vapor pressure until all the molecules had sufficient speed to escape when the liquid would explode. (This assumes probably that only a fairly thin layer of liquid is present, as this would not hold good for a deep layer of liquid.) In this paper the author attempts to determine the numerical values of the various factors involved in the evaporation of liquids and does this by graphical methods. He assumes that the ratio of the density of the vapor above the liquid to the density of the liquid is really the ratio of the number of the molecules with sufficient speed to escape from the liquid to the number with insufficient speed. Thus, if D_v and D_e are the densities of the vapor and liquid, respectively, then $D_v/D_e = N/N_i$ where N is the number of molecules with sufficient speed to escape and N_i the number with insufficient speed.

In a previous paper by Ramsay and Shields (Philosophical Transactions, 1893, vol. 184, p. 647) the densities of both liquid and vapor are given from ordinary temperatures up to the critical temperature for a number of materials and the author uses these data.

By using graphical methods and plotting a curve on a scale of 3 ft. by 2 ft., he first derives the value of x corresponding to the value of D_v/D_e, which is the minimum velocity at any temperature which a molecule must have to escape from the liquid. This velocity will, of course, be relative to the most probable velocity at the temperature considered taken as unity. This gives a table in which one of the columns gives a minimum velocity relative to the most probable velocity at the temperature which a molecule must have to escape from the liquid. At the critical temperature, where the densities of the liquid and the vapor are the same, and where there is no force to be overcome by a molecule in passing from one layer to the next, there will be no definite minimum velocity, but each molecule may be considered free to pass any imaginary plane provided it is in a suitable position and has a velocity component in the right direction. Thus, at this temperature, a molecule with a velocity less than the mean might pass an imaginary plane, provided it was nearer to that plane than its average mean free path and its velocity was perpendicular to the plane. When, however, a definite force has to be overcome to escape from the liquid phase to the gaseous phase, then only those molecules with a velocity greater than a certain minimum are able to escape.

Another column in the same table gives numbers proportional to the minimum kinetic energy of the escaping molecules, and expressed by Mx^2, where M is the molecular weight of the substance in the gaseous state. In this column the minimum velocity x, instead of being expressed relative to the most probable velocity at the temperature considered, is expressed relative to the most probable velocity at 20 deg. cent., which is taken as unity. The velocities at other temperatures have also been calculated.

Next, the minimum kinetic energies of escape are plotted against the temperatures and this shows that for the unassociated substances, such as ethyl oxide, carbon tetrachloride, etc., the minimum kinetic energies of escape fall off linearly with the temperature to within 20 deg. or 30 deg. of the critical temperature. They then decrease rapidly to zero. The curves for acetic acid and the alcohols are, however, more complicated.

In order to escape from the liquid, a molecule must have sufficient energy to overcome the molecular surface energy of the liquid and after that to expand to the vapor. The work done in expansion is given by $p(V_v - V_e)$ where V_e and V_v are the specific volumes of the liquid and vapor, respectively. Except near the critical temperature, when V_e becomes comparable to V_v, this becomes $pV_v = RT$. That is, the external work is proportional to the absolute temperature, except near the critical temperature. The other work which the molecule has to do is overcoming the molecular surface energy. This falls off linearly with the temperature up to within 20 or 30 deg. cent. of the critical temperature. It is to be expected, therefore, that the minimum energy of escape should vary linearly with the temperature. The marked decrease in the minimum energy close to the critical temperature is explained by the marked decrease in the forces to be overcome, as shown by the rapid decrease in the latent heat of vaporization near the critical temperature.

It has been pointed out that up to within 30 or 40 deg. cent. of the critical temperature the external work can be expressed by $PV = RT$. At a given temperature this will be approximately the

same for all liquids. The internal work will, however, differ according to the molecular surface energy.

From a table giving the minimum energy of escape and the molecular surface energy of a number of substances at 150 deg. cent. and from a graph made from this table, it appears that the unassociated substances fall practically on a straight line, but the definitely associated bodies fall away from the line. The relationship between the unassociated substances would appear to indicate that the minimum energies calculated stand for a definite physical reality, and the author applies these ideas to a solution of the problem of the latent heat of vaporization.

According to the view of the kinetic theory of evaporation, which is the basis of this paper, it is the more rapidly moving molecules which escape. These, it seems, must have a minimum energy before they can leave the liquid and become incorporated in the vapor. They must have sufficient kinetic energy to enable them to overcome the cohesive forces of the liquid and after that to expand. In their final condition their mean kinetic energy is the same as the mean kinetic energy of the liquid as a whole, being at the same temperature. The difference, therefore, between the total kinetic energy of one gram of the escaping molecules and the total kinetic energy of one gram of the liquid or vapor as a whole, must be the latent heat of vaporization of the liquid. In order, therefore, to find the latent heat of vaporization it is necessary to find the difference between the total kinetic energy of one gram of molecules having a certain minimum velocity and the total kinetic energy of one gram of the general mass of molecules.

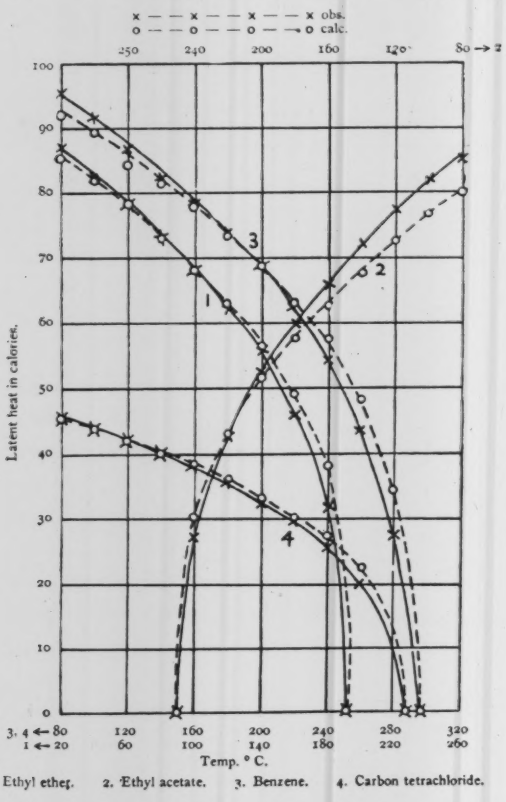


FIG. 16 LATENT HEATS OF VAPORIZATION AS CALCULATED AND OBSERVED

This can be done by integration or graphically, and the author uses the latter method. It was found that when certain changes in the estimated velocity of the molecule have been introduced, not only are the actual heats of vaporization of the different unassociated substances obtained in their right order of magnitude, but that they are all correct among themselves and over the range of temperature up to the critical temperature to within a few per cent of error, and that when allowance is made for certain factors the observed and calculated values are almost identical. In order to show the closeness of the shape of the calculated and observed curves, the values for four substances have been plotted in Fig. 16.

In this calculation, the latent heat of vaporization has been definitely related to the total kinetic energy in one gram of the substance at the temperature considered. As the kinetic energy associated with each molecule at the same temperature is the same, it follows that as there are fewer molecules in one gram of a substance at high molecular weight, there is also less total kinetic energy. The latent heat of vaporization is, therefore, also less for substances of high molecular weight than for these of low molecular weight. This is supported by reference to a table of latent heats of vaporization for a number of substances.

The fact that the latent heat of vaporization can be calculated in this way, from first principles, is a strong proof of the truth of the initial assumption, namely, that the ratio of the density of the vapor to the density of the liquid is the ratio of the number of molecules with sufficient speed to escape to the number with insufficient speed.

A simple explanation of many of the problems of vapor pressure suggests itself from this fact. Thus the lowering of the vapor pressure of a solvent by the introduction of non-volatile matter is explained by saying that the number of molecules with sufficient speed to escape is thereby reduced relatively to the number of insufficient speed. On account of their electrical nature, ions have probably to overcome a great cohesive force in order to escape and consequently need a much greater kinetic energy than ordinary water molecules. They are, therefore, largely non-volatile. In a similar way the forces of attraction between water molecules and other non-volatile molecules is probably greater than that between the water molecules.

As the laws of osmotic pressure are very similar to those of vapor pressure, it is extremely likely that, in osmotic pressure, only those molecules with a speed greater than a certain minimum are able to penetrate the semi-permeable membrane. The membrane itself is probably very similar to the surface of a liquid, this is, in effect, a semi-permeable membrane permitting only those molecules with a certain velocity to pass through. Molecules are, probably, unable to penetrate the membrane for the same reason that they are non-volatile, namely, the forces of cohesion on them are too great to enable them to break away, with the speeds they possess, from the solution in which they are. The introduction of a number of molecules unable to penetrate the membrane, reduces the number on one side relative to the other which can penetrate the membrane and gives rise to the ordinary phenomena of osmotic pressure.

It is not unlikely that some of the facts in connection with the latent heat of fusion would be explainable in a similar way to those in connection with the latent heat of vaporization. It is not proposed at present, however, to attempt such an explanation. (Dr. D. B. Macleod. *Transactions of the Faraday Society*, vol. 20, pp. 3, April, 1925, pp. 525–543, 8 figs., *tA*)

TRANSPORTATION

Railplane System of Transport

A NEW system of transport has been devised by George Bennie, Bute, N. B., in which the cars are suspended from a rigid overhead structure and are propelled by means of airscrews. The utility of the system rests in the possibilities which it opens up of affording a rapid transport for passengers, mail, newspapers, perishable goods, etc. The cars are shaped somewhat like airships. In order to reduce friction to a minimum, a single overhead rail and modern ball-bearing devices are proposed. The design of the cars is such that at high speeds they tend to rise slightly in the air and thus relieve to a great extent the friction due to the weight of the car on the rail. The design of the suspension brackets between the bogies overhead and the car beneath is such as to check the tendency of the car to rise in the air beyond the amount which is required for relieving the weight on the trucks. (*The Railway Gazette*, vol. 43, no. 13, Sept. 25, 1925, pp. 381–382, illustrated, *d*)

CLASSIFICATION OF ARTICLES

Articles appearing in the Survey are classified as *c* comparative; *d* descriptive; *e* experimental; *g* general; *h* historical; *m* mathematical; *p* practical; *s* statistical; *t* theoretical. Articles of especial merit are rated *A* by the reviewer. Opinions expressed are those of the reviewer, not of the Society.

Engineering and Industrial Standardization

International Standard Power-Test Codes

A NUMBER of years ago the International Electrotechnical Commission, now constituted by representatives of twenty-two countries, organized an Advisory Committee on Prime Movers for Electrical Plant. While to date this committee has not been very active in this field, it completed a report on Nomenclature of Hydraulic Turbines, which was published by the I.E.C. in July, 1914, as Publication No. 29.

New impetus to activity in this field by the I.E.C. was supplied by a resolution drafted and endorsed by the delegates to the World Power Conference at Wembley, London, July, 1924. The import of this resolution was to request the International Electrotechnical Commission to undertake in an energetic way the development of a series of codes or specifications for the testing of steam engines, steam turbines, and hydraulic turbines, as well as oil and gas engines. In requesting the I.E.C. to undertake this task the representatives of the power-producing industry expressed the hope that the machinery established and the experience so far obtained in the development of international electrical standards might be put to good use in the formulation of a set of international power-test codes.

Recent informal conferences on this subject indicate that when finally completed each of these international codes will be written in the form of an acceptance test of the prime mover in question and probably will consist of at least the following five sections: (1) Nomenclature, (2) Units and Symbols, (3) Important Measurements, (4) Instruments and Apparatus, their types and limitations, and (5) Data and Results, including some essential computation methods.

At the request of the U. S. National Committee of the I.E.C., The American Society of Mechanical Engineers has accepted membership on this national committee and has appointed Dr. Fred R. Low and Dr. William F. Durand as its two representatives on the Committee and Mr. C. Harold Berry as their alternate. The U. S. National Committee recognizes the Society's long experience in the formulation of power test codes and has cordially welcomed its offer to coöperate in this important and timely international undertaking.

American Standards for Shafting and Keys Ready for Distribution

TWO important dimensional standards dealing with cold-finished shafting, and square and flat shafting keys, approved as Tentative American Standards by the American Engineering Standards Committee, have recently been published by The American Society of Mechanical Engineers, the sponsor, and are ready for distribution.

The sizes covered are from $^1/_2$ to 6 in. (machinery shafting) and $^{15}/_{16}$ to $5^{15}/_{16}$ in. (transmission shafting). The recommended stock lengths for cold-finished shafting are 16, 20, and 24 ft. All the tolerances are negative, and represent the maximum allowable variation below the exact nominal size.

The keys which have been considered for these sixty standard shaft diameters are either square or flat. The standard widths and heights and the corresponding negative tolerances are given. The keys are to be cut from cold-finished stock and are to be used without machining.

In the development of these standards, which was begun in 1918, the whole industry, including not only manufacturers and users but more than 300 dealers and jobbers, has participated by correspondence, conferences, and representation on the sectional committee in charge. A tentative report published in 1923 was widely distributed. As a result of this action, a revised report was again widely circulated for criticism before final revisions were made.

The Sectional Committee is now working on formulas to guide engineers in the selection of the best sizes for transmission shafting for use under various conditions for loading. The chairman of the committee is Cloyd M. Chapman, and its secretary is C. B. LePage.

The standards for shafting and keys, hereafter to be known as B 17a and B 17b, respectively, are the *first* dimensional standards having national approval to be published in this country in the single-sheet form. They may be secured from the A.S.M.E. and the A.E.S.C., 29 West 39th Street, New York, N. Y., at a price of 20 cents per sheet, with a reduction for orders in quantity.

Taper Shanks Up for Standardization in Sweden

STANDARDS for tool tapers are not entirely adequate to the requirements of modern manufacturing and trade conditions, and no general agreement has been reached as yet among the various countries. The Swedish National Standards Committee have proposed that the Brown & Sharpe tapers, up to and including No. 16, should be adopted for milling machines, and that the Morse tapers, extended according to a plan proposed by Germany, should be used for drilling machines.

The whole subject is being actively discussed in Sweden and in one or two other continental countries, and will probably move along actively as the general criticism of the drafts have been studied.

This is one of the four pieces of standardization on the program of the Sectional Committee on the Standardization of Small Tools and Machine-Tool Elements, Harry E. Harris, Chairman of Central Committee. The other three are Tee Slots, Tool Holders and Tool-Post Openings, and Spindle Noses. The Sub-Committee on Tool Holders and Tool-Post Openings is now being organized, and good progress has already been made by the Sub-Committee on Tee Slots.

Fire-Hose Coupling Screw Threads

A SUSTAINING-Membership Bulletin recently issued by the A.E.S.C. states that a note from the Bureau of Standards refers to the work of rethreading to the new American (National) Standard of all the fire hydrants, hose couplings, and special connections in the District of Columbia. The Bureau has been assisting in this work, having loaned to those in immediate charge of the work the set of salvaging or rethreading tools which was given to it by the National Board of Fire Underwriters.

Report on Wrench-Head Bolts and Nuts and Wrench Openings

A NEW draft of the proposed Tentative American Standard for Wrench-Head Bolts and Nuts and Wrench Openings has been prepared by Sub-Committee No. 2 of the Sectional Committee on the Standardization of Bolt, Nut, and Rivet Proportions, under the sponsorship of The American Society of Mechanical Engineers and the Society of Automotive Engineers. The tables of sizes given are intended to supersede all existing standards of wrench-head bolts and nuts which have grown up in the various industries. The work reflects the tendency of recent years toward the more economical use of material shown in smaller head and nut dimensions as typified by the present S.A.E. standards. In all cases nominal widths across flats of bolt heads and nuts have been taken as maximum sizes, so that manufacturing tolerances are all minus. The minimum dimension of wrench opening has been made such as to provide a clearance between the maximum nut and the minimum wrench, so that both manufacturing limits of a wrench opening are above nominal size. The tables given cover finished and semi-finished square and hexagonal bolt heads; finished regular square and hexagonal nuts; finished jam nuts; rough and semi-finished

regular square and hexagonal bolt heads; rough regular and semi-finished square and hexagonal nuts; special light nuts, fine-thread series; hexagonal and square machine-screw and stove-bolt nuts; set-screw heads; finished hexagonal cap-screw heads; and open-end wrench openings.

The draft of this standard is now before the Sectional Committee in page-proof form. After the letter ballot of the Sectional Committee has been completed, the proposal will be formally submitted to the sponsor bodies, who will in time transmit it to the American Engineering Standards Committee for its approval as Tentative American Standard.

Abbreviations and Symbols Committee Organized

ORGANIZATION of the Sectional Committee on Scientific and Engineering Symbols and Abbreviations is almost completed, and it is expected that a meeting will shortly be held to organize sub-committees for the detailed technical work. While it was originally intended that the work to be taken up by the Sectional Committee should include a broad program of unification of graphical symbols and symbols for quantities in equations and formulas, the present plan is to concentrate upon a few very definite subjects in which sufficient work has already been done by different organizations to make the subjects ripe for immediate unification—for example, fundamental symbols in mechanics, in electrical engineering, civil engineering, heat engineering, illumination, and aeronautics. Sponsors of the project include the Society for the Promotion of Engineering Education, American Association for the Advancement of Science, American Institute of Electrical Engineers, American Society of Civil Engineers, and The American Society of Mechanical Engineers.

Education in Standardization in German Technical Schools

GERMANY has realized the importance of introducing to engineers during their student days a knowledge of the fundamentals of standardization, according to a representative of that country's standardizing body who recently visited here. The technical schools are among the largest subscribers to standard sheets as they are issued by the Normenausschuss der Deutschen Industrie. Special reductions in price are given to students to encourage the purchase of whole sets of standards, and arrangements are made by the N.D.I. with a particular instructor in each technical school by which he is designated the special purveyor of knowledge on the standardization movement and its application. For this purpose he receives a complimentary set of standards and papers relative to standardization, together with motion-picture films to be used in further education along these lines.

Thus is laid the background for a thorough and practical understanding of the principles of standardization for the young men who enter German industry. The technical college at Dresden, Germany, has established a special course of instruction in technical measurements and the principles of manufacture of interchangeable products. This course will deal with the entire field of standardization as a necessary part of the groundwork of student engineers taking such a course. Dr. Berndt, an eminent specialist on screw threads and on mass-production problems, will be in charge of this work. The establishment of such a course in training may be said to be rather typical of the latest developments along educational lines in Germany.

The standard sheets issued by the N.D.I. are regularly used for instruction purposes in machine design and in other fields in which standardization has taken place. Such use, it will be readily seen, puts the student in direct touch with actual developments in industrial practice, which means that the transition from the training school to his place in the industrial world is made more natural and more satisfactory, both to his new employer and to himself. Information obtained in the technical school can be applied by the young worker without hesitation as recognized and acceptable, and the embarrassment and expense of "breaking in" the young college graduate is spared.

Out of an edition of 15,000 copies of the "Dinbuch" on technical drawings which was sold in a few months, the first lot issued, about 4000 in number, was sold chiefly to students in technical schools.

To be on a self-supporting basis through the sale of its standards is one of the present aims of the N.D.I. The time and money now expended on each project is borne by the trade association sponsoring it, even though the work may have been carried on by the N.D.I. engineers. It will probably not be long, however, before the German standards committee will be able to realize its desire to be at least partly self-supporting from their sale of standards. There are about 1200 regular subscribers to all of the sheets issued, and between 600 and 800 subscribers to the sheets dealing with special branches. The first edition of each sheet consists of 4000 copies, and others follow as each is exhausted. The first and second editions, for instance, of the standard sheet on screws with fillister heads have been sold and the third edition has now been put on sale, which means that more than 8000 copies of this particular standard have been distributed. One large firm regularly takes 800 copies of each sheet issued.

The symbol "DIN" has been made a legal trade mark by the standards committee, and is used as extensively and as conspicuously as possible. For example, each year the N.D.I. has a booth covered with DIN placards, at the well-known Leipzic Exposition, distributing information on standardization, issuing posters to people who supply standard articles, and generally carrying on an educational program along the lines of standardization.

Coöperation between industry and the national standardizing body is very close. The staff of the N.D.I., 50 people in all, includes 13 engineers, each of whom handles definite projects either in actually carrying on the work or in an advisory capacity, correlating his efforts with those of the engineers employed by individual firms which have standardization departments. At least a hundred firms have standardization departments of their own which employ as many as seven or eight engineers. One of the advantages of this system is that the N.D.I., keeping in close touch with these private factory standardizing bodies, has every facility for having experiments made with regard to questions arising in connection with establishing standard practice.—From the A.E.S.C. Sustaining-Membership Bulletin of June 30, 1925.

A.E.S.C. Financial Statement

THE American Engineering Standards Committee through its Secretary, Dr. P. G. Agnew, has submitted to the members of the A.E.S.C. Finance Committee the following financial statement for the period January–June, 1925:

Balance January 1, 1925		$ 5,467.03
RECEIPTS		
Dues	$ 9,750.00	
Sustaining memberships	16,028.41	
Sales of standards:		
Domestic	184.81	
Foreign	12.15	
Contributions	5.00	
Miscellaneous	45.55	
Miscellaneous refunds	692.14	
Interest on bank balances	59.33	
Total receipts		26,777.42
Total		$32,244.42

DISBURSEMENTS

	Net expenditures (disbursements less refunds) Jan.–June, 1925	Budget Jan.–June, 1925	Budget July–Dec., 1925
Salaries	$18,370.86	$17,695.86	$17,500	$19,000
Travel	250.96	250.96	700	600
Rent	1,791.00	1,791.00	1,800	1,800
Office supplies	1,305.69	1,305.69	1,200	1,200
Postage	1,243.79	1,243.79	1,200	800
Mimeographing	1,088.66	1,088.66	1,000	1,200
Printing	1,200.00	1,200.00	1,500	400
Telephone and telegraph	269.28	269.28	300	300
Office equipment	642.90	642.90	700	500
Purchase of standards:				
Domestic	353.92	169.11	600	700
Foreign	35.30	23.15		
Miscellaneous	771.58	771.58		
S.M. campaign	.....	.....	....	1,000
Total	$27,323.94	$26,451.98	$26,500	$27,500
Bank balance June 30	3,922.49			
On hand for deposit	997.99			
	$32,244.42			

Correspondence

CONTRIBUTIONS to the Correspondence Department of Mechanical Engineering are solicited. Contributions particularly welcomed are discussions of papers published in this journal, brief articles of current interest to mechanical engineers, or comments from members of The American Society of Mechanical Engineers on articles or policies of the Society in Research and Standardization.

Critical Bearing Pressures

TO THE EDITOR:

In the paper by L. N. Linsley[1] on critical bearing pressures causing rupture in lubricating-oil films, quotation is made of Sommerfeld's, Nicolson's, and Moore's equations for critical or breakdown pressures. No comment is made on the comparison of these equations, which differ even in form, so that if one of them is true the others cannot be. In Mr. Linsley's notation Sommerfeld's equation is $P = \frac{\mu N}{349557}\left(\frac{d}{\Delta}\right)^2$, where P is pressure per square inch of projected area, N revolutions per minute, Δ the radial clearance between shaft and bearing (*not* oil film thickness, necessarily), d the diameter of shaft in inches, and μ the absolute viscosity in poises. This equation is for a complete bearing.

Nicolson's equation is $P = 40(dN)^{1/4}$. Moore's is $P = 7.47\sqrt{V}$, where V = rubbing velocity in feet per minute. Since $V = \pi dN/12$, this last is equivalent to $P = 3.82(dN)^{1/2}$.

Nicolson's and Moore's equations do not recognize the viscosity of the oil nor the clearance ratio of the bearing as being concerned; nor do they involve even the diameter of the bearing correctly in regard to the speed. Sommerfeld's equation shows that N and d are not concerned to the same exponent in the equation for P. Rubbing velocity, or surface speed of the shaft, can be involved as such in critical pressure conditions of a bearing only when d and N have the same exponent in the equation for P. This was assumed by Moore; it is a notion which has been and is widely prevalent—but it is not true, theoretically or experimentally.

A considerable body of experimental data on critical conditions in complete bearings has been gathered and analyzed recently in papers by Wilson and Barnard before the Society of Automotive Engineers. Working with moderate pressures, and causing failure by lowering speed or viscosity or both, they find that bearings have their critical conditions (minimum coefficient of friction, on edge of failure) at a characteristic value of $\mu N/P$; and this critical value of $\mu N/P$ is a function of the clearance ratio Δ/d of the bearing. Rather interestingly, Wilson and Barnard find the critical value of $\mu N/P$ was little affected by the "body" of the oil, although it was measurably lower for an oil of higher body and same viscosity.

Moore worked with a half-bearing instead of a complete one. His combination of dN, or assumption that d and N are equally involved in the critical conditions through a wide range of bearing designs, would almost certainly have been given up as impossible if he had worked with a number of bearings of various combinations of diameter and clearance ratio. In any case Moore's numerical coefficient would not be correct for a complete bearing. The wide acceptance of his equation is difficult to understand.

If bearings of various sizes were geometrically similar (had the same clearance ratios), the critical value of $\mu N/P$ would be independent of d. But a study of a series of "similar" bearings, as actually made—for example, a Westinghouse series of sizes of electric motor bearings—shows that in practice Δ is not proportional to d as d varies, but more closely varies as $\sqrt{d}$. Hence d/Δ is not the same for bearings *described* as similar, but varies about as constant $\times \sqrt{d}$. Putting this into Sommerfeld's equations gives P = constant $\times Nd$. With this is to be tied the general tendency for N to be lower as d is larger, in a series of "similar" machines of varying size; and the drop of μ with increase of Nd because of temperature rise. But since we now design machinery to definite values of Δ as well as of d and N, we might as well use the Sommerfeld style of equation as a design equation.

There is a technical mistake in the design of Mr. Linsley's testing machine. He was testing two half-bearings opposed to each other diametrally across the shaft for ease of load application. A half-bearing run by itself naturally moves sidewise with regard to the shaft so that the thickness of oil film at the entering edge is different from that at the leaving edge. Such sidewise motion of Mr. Linsley's two half-bearings was prevented by their construction with dowel pins running from one into the other. The effect of this improper constraint on his experimental results is impossible to predict in magnitude, but it must surely have made his critical pressures lower than they would have been had the half-bearings been free to move sidewise with regard to each other and the shaft. His difficulty in wearing in his bearings may also have come in part from this improper constraint. This difficulty of wearing in, however, is always present and is the main reason why it is experimentally so difficult to investigate the effect of clearance ratio on the characteristics of a bearing.

Mr. Linsley's experimental procedure in finding the critical pressures by raising the pressures to high values, while keeping up the speed and viscosity, is novel and important; and ultimately this procedure explains why his results are so different from previous determinations of $\mu N/P$ at the critical conditions, made by decreasing μ or N or both while keeping P values fixed and moderate.

Mr. Linsley's final tests were made with two different mineral oils, of Saybolt viscosities 308 and 205 seconds at 100 deg. fahr. (absolute viscosities about 0.6 and 0.4 poises at 100 deg. fahr.). The two gave substantially identical results, when their viscosities are taken into account. In further tests these oils were modified by the addition of 2 per cent oleic acid (to increase their "body") with the interesting result that the critical pressures turned out to be lower instead of higher for the same speeds and viscosities!

These final tests, with the untreated and treated mineral oils, Mr. Linsley summarizes in two figures of breakdown pressure P vs. rubbing velocity V, with points representing tests, and families of curves drawn in on lines of constant viscosities. These viscosities are subject to an unknown systematic error because they were found by taking oil temperatures at the edges of the half-bearings instead of in the oil films; but this is probably a minor error which cannot change greatly the character of the results. Cross-plotting of the curves in Mr. Linsley's figures shows that P varies closely as $V^{1/4}$ (or $N^{1/4}$) for constant μ and as $\mu^{1/4}$ for constant V (or N). Hence the writer calculated the $V\mu$ products from the tables of original data, and plotted P vs. $V\mu$ of all the tests, 39 with untreated mineral oil No. 5, and 36 with untreated mineral oil No. 1, finding all but a few low points to fall upon a single curve of empirical equation $P = 1315(V\mu)^{1/4} = 1120(N\mu)^{1/4}$.

Apparently Mr. Linsley's results check in form, although not numerically, with Nicolson's equation rather than those of Moore or Sommerfeld. The equation $P = 1120(N\mu)^{1/4}$, which fits Mr. Linsley's results, may be put in the form $N\mu/P = P^3/1120^4$. Sommerfeld's equation is $N\mu/P = 349557(\Delta/d)^2$. Supposing failure to occur at a characteristic value of $N\mu/P$, as Wilson and Barnard have found, and equaling $349557(\Delta/d)^2$ with $P^3/1120^4$ there results a calculation of Δ as a function of P in Mr. Linlsey's bearings as follows:

P	2500	3000	3500	4000	4500	Load per sq. in.
Δ	0.00034	0.00044	0.00056	0.00068	0.00082	Radial clearance

Under the high loads of Mr. Linsley's tests, deflections of the bearing by the load and by uneven heating might be expected; and these calculated values of Δ plot as a very fair linear relation to P! One cannot say, then, that Sommerfeld's theory is disproved; only that in practice close-fitted bearings running under high loads will experience distortions large compared to the cold clearance of the bearing, and which must be considered in predicting its capacity.

[1] Trans. A.S.M.E., vol. 46 (1924), p. 855.

Unknown factors in the problem, which should be taken into account in future work, are: (1) the radial clearance, measured as under running conditions of temperature and temperature distribution; (2) the real temperature of the oil in the lubricating film; (3) the maximum pressures and the approximate pressure distributions in the lubricating film; (4) the shape distortion of bearing and shaft under the applied loads. (2) and (3) are needed in order to know the viscosity of the lubricant, for under unit loads as high as in Mr. Linsley's experiments the pressure effect on viscosity of mineral oils may not be negligible. On the other hand, the temperatures in the oil film where the film is very thin are very much higher than the temperatures of the bearing metal or than the temperatures of oil at the edges of the bearing, and this tends to lower the viscosities. Further, Mr. Linsley's results, combined with those of Wilson and Barnard, show that "body," as a property of a lubricating oil, is of little importance or even a detriment to the lubrication of a shaft rotating in a bearing—viscosity is the important property in this case.

G. B. UPTON.[1]

Ithaca, N. Y.

Concerning Supersaturation

TO THE EDITOR:

Undoubtedly, according to the kinetic theory of gases, when a substance undergoes sudden expansion, a certain time interval must elapse for readjustment of the energy distribution. Consequently undercooling necessarily results.

Unfortunately, direct experimental verification of undercooling in actual turbines seems to be impossible with the present known methods of observation, and so far, the Westinghouse Company has not been able to detect any action in its turbines to indicate undercooling, except that it has been noticed in one or two cases that the slight erosion of the blade tips in the reaction blading has not appeared at the point where it would occur if the steam were in thermal equilibrium, but rather at a point lower down, where the undercooling theory would indicate that it should. Such indications are too indefinite to be given much weight.

Because the steam tables now used are not worked out for supersaturation, and because of the much greater difficulty of calculations, the Westinghouse Company does not take undercooling into consideration in the design of turbines, as it is felt that the effect on efficiency would not warrant the extra work involved.

In regard to nozzle coefficients: In actual turbines as built, with nozzles having a more or less sharp bend just before the throat, the reported high nozzle coefficients have not been observed, the most reliable tests indicating nozzle-flow coefficients of from 96 to 98½ per cent. Where coefficients of unity or above have been found, an examination of the nozzles has invariably shown leakage, which, when corrected, reduces the coefficient of flow to about the figures previously given.

The Westinghouse Company for many years has used nozzle data which were calculated from the kinetic theory of gases without any reference to experimental work. No claim is made for the correctness of the data. It is remarkable, however, that they have agreed with the results obtained from turbines under nearly every conceivable condition of design and expansion range, and under conditions which other available data would not check.

Briefly stated, this theory is based on the assumption that, *during acceleration only*, there is a reduction of the relative velocity and the number of collisions of the molecules in the direction of acceleration, with the result that part of the molecular energy becomes "locked up" and is not available to cause further increase in velocity. The effect of this, down to the critical pressure, is very slight, but it is very considerable after the critical velocity is reached.

Possibly this line of thought may suggest a new method of attack to some engineer or physicist to further develop the theory of flow in nozzles.

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Caustic Embrittlement of Boiler Plate

THE phenomenon of crystallization and cracking of boiler plate at or near the seams—due not to corrosion or wasting away of the plate but rather to some very different action—has been observed and known for upward of 30 years. The effect was first noticeable in the unexplainable leakage at the joints of certain boilers, followed in certain instances by explosions.

During the past 10 or 15 years more pronounced examples of this trouble have arisen which have indicated that it may under certain circumstances prove a dire menace to the safety of boilers.

In 1912, difficulties of this character experienced with the boilers in the power plant at the University of Illinois at Urbana led to an investigation of the phenomenon from both chemical and physical viewpoints.

The experiments were directed toward determining the effect of caustic soda upon steel or the indirect effect of hydrogen resulting from such action. In general there seemed to be sufficient data for concluding that:

a Caustic soda of sufficient strength attacks iron with the generation of hydrogen. Indicating the reaction with the hydroxyl ion only, we would have $3Fe + 4OH = Fe_3O_4 + 4H$.

b Hydrogen in the nascent state, whether generated by alkali or acid, enters into the texture of the iron in a way to modify its physical properties.

c The hydrogen effect, at least in its first application, is transient, and after a period of rest or freedom from the hydrogen action, the iron reverts to its normal condition.

d Sodium carbonate is without action on iron; there is, therefore, no generation of hydrogen. The hydrolysis of sodium carbonate is directly dependent upon the temperature maintained, the withdrawal of the vapor of CO_2, and the admission through the feedwater of carbonated water. It is evident, therefore, that the chemical activity would vary with the ratio of hydrolization or the degree of concentration of the sodium hydroxide.

d Certain accompanying salts, as the chromates, have an inhibitive effect. Other salts, as sulphates, have at least the effect of acting as diluents. The limits of concentration for maintaining a condition which would be below the danger point have not been studied, but are features of the case which are of the utmost importance.

Another important investigation of this abnormal type of failure in mild steel characterized by intercrystalline cracking has for some time been under way at the Massachusetts Institute of Technology, and Dr. R. S. Williams, Associate Professor, and V. O. Homerberg, Instructor in Metallography, have reported upon their investigations in a paper before the Boston Chapter of the American Society for Steel Treating, as follows:

The results obtained from the tests lead to the following conclusions with regard to the intercrystalline cracking of steel under the influence of caustic solutions:

1 During the crystallization of steel, the impurities, to a considerable extent, are rejected to the grain boundaries.

2 The oxides and sulphides are two of the prime factors in caustic embrittlement.

3 The oxides are reduced under the influence of cathodic hydrogen.

4 The sulphides are removed due to the action of hot caustic soda solutions.

5 The removal of the sulphides produces a surface condition favorable to progressive corrosion.

6 Assuming that progressive corrosion starts with the removal of the sulphides, the corrosion will be greatly accelerated if the material is stressed. Furthermore, when the steel is under tension there is a tendency for the matrix to pull away from the inclusions at the grain boundaries and in this manner to produce small capillaries into which the corroding solution can penetrate.

7 In stressed areas containing oxides, the volume increase due to the reaction with cathodic hydrogen, may produce stresses which added to those initially present, may cause cracking.

8 It seems evident that hydrogen acts in three ways to produce embrittlement; *first*, the temporary brittleness caused by absorbed hydrogen (as in acid pickling); *second*, it acts to reduce oxides; and *third*, its effect due to the change in volume at the grain boundaries, resulting because of the production of water. This latter volume increase would create a stress which, added to those originally present, may cause cracking, especially at those points where these stresses are at a maximum.

The Boiler Code Committee of The American Society of Mechanical Engineers has given the problem some consideration, but in view of the lack of authentic data concerning the phenomenon and the incomplete condition of the various investigations now under way upon it, it has been found impossible to dispose of the question with any definite recommendation as yet.—C. W. Obert, C. W. Edgerton and A. G. Pratt, in a recent report to the president of the American Boiler Manufacturers' Association.

Centerless Grinding

"Through Feed" and "In Feed" Methods of Centerless Grinding—Increased Production of Pistons and Piston Pins Obtained by Use of The Centerless Grinder—New Theory on Relation between Control Wheel, Cutting Wheel, and Work

THE paper by W. J. Peets on The Theory and Practice of Centerless Grinding which was presented on September 8, 1925, under the auspices of the A.S.M.E. Machine Shop Practice Division, at a session of the New Haven Machine Tool Exhibition, elicited much interesting discussion and emphasized the importance that this process of grinding is attaining in interchangeable manufacture. Mr. Peets' paper appeared in the September issue of MECHANICAL ENGINEERING. The discussion thereon follows.

W. W. Seabury[1] submitted a written discussion in which he said that in the production of cylindrical parts the centerless grinder represented a radical departure from the stereotyped method of holding the work between centers during the machining operation.

The centerless grinder was the result of a gradual development of machining skill, fulfilling the demand for a precision finish of cylindrical parts with a minimum of cost. Generally speaking, the outstanding advantages of this type of machine were the increase in production obtainable at reduced cost and the greater precision obtainable. This was due to the fact that the process of grinding with the centerless grinder was continuous. Idle time lost in loading and adjusting the grinding wheel of a center-type grinder for each part to be ground was eliminated. Extra operations such as centering, inspecting, and removing each part from the machine were dispensed with in the centerless machine.

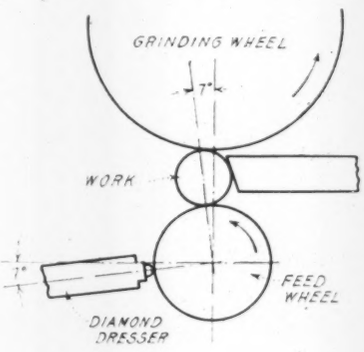

FIG. 1

The general opinion among many engineers and production men was that the scope of the centerless grinder was limited to the production of small cylindrical parts ranging from 1/16 in. to 3 in. in diameter; and that it was primarily efficient as a finishing machine and not as fast in machining cast iron and steel parts accurately as the various special turning machines on the market. His experience had proved, however, that the range of work was very wide.

As distinguished from the center type of machine, the centerless machine comprised two wheels, a high-speed grinding wheel and an opposed regulating wheel revolving slowly opposite the high-speed wheel and forming a grinding throat as shown in Fig. 1. The part to be finished was fed into the throat of the wheels by gravity and was supported by a work rest. The grinding wheel served entirely for grinding purposes; the regulating wheel had two functions: first, to control the speed of rotation of the work, and second, to regulate the feeding movement. With a hopper feed which brought the work to rest, and the regulating wheel feeding it through the machine, the operation of the centerless grinder becomes automatic.

The capacity of the machine depended upon several factors: (1) The angular adjustment of the regulating wheel, or the angle of its axis to that of the grinding wheel; (2) the abrasive capacity and speed of the grinding wheel; (3) the amount of stock to be removed; (4) the finish required; (5) the type of work to be ground, whether solid or hollow and having thin walls.

There were two primary methods of grinding the work with the centerless grinder, wrote Mr. Seabury, each of which had its advantages and was adaptable to a certain form of work. One was the "through feed" method, adaptable to straight cylindrical work so-called because it passed axially through the machine between the two wheels. The other, the "in feed" method, was used on parts having a shoulder or taper. In the latter case the parts were fed into the grinding throat either vertically or laterally and supported or steadied during the grinding, and were subsequently ejected.

Some of the best examples of the "through feed" method in his company's production were the Model "T" and tractor piston pins and pistons and the brake shoe. An example of the "in feed" method was the push rod.

Before the introduction of the centerless grinder it had required 43 machines and approximately 130 men to produce 50,000 Model "T" piston pins per 24-hour day, or 384 finished pins per man per day. With the centerless grinder the production had jumped to 66,000 pins with a complement of 157 men operating 26 machines, or a production increase of 36 pins per man and a 10 per cent reduction in cost, to say nothing of the further reduced cost resulting from the decrease in maintenance per machine and the fewer machines required, also the minimizing of investment in machinery and floor space required. Another pertinent factor in cutting the total cost 60 per cent had been the reduction in scrap of over 30 per cent and the greater accuracy obtained. Limits had been reduced to 0.0004 in. for roundness and taper.

The grinding of the tractor piston pin was another part where greater accuracy, increased production, and reduced cost had been obtained with the "through feed" centerless grinder. The grinding operation of the centerless machine being done on the three-point-contact method as distinguished from the one-point contact with the center-type grinder, accounted for the fast cutting action of the machine and accuracy of the work. In the centerless machine the work was supported directly opposite the grinding wheel by the regulating wheel and also at another point opposite by the rest, as might be clearly seen from the illustration. Some idea of the accuracy obtained might be had from the fact that a pin did not pass inspection if it was out 0.0002 in. in roundness or taper. On this pin 1 3/8 in. in diameter, 0.018 in. to 0.020 in. was removed from the diameter in three passes in the roughing. About 0.0035 in. was removed in five finishing operations as follows:

1st pass	0.0012	4th pass	0.0005
2nd pass	0.0008	5th pass	0.0003
3rd pass	0.0007	Total	0.0035

In order to detect any inaccuracy in the piston pins at inspection, the gaging was done by a method using the three-point principle employed in the centerless grinder. A V-block and magnifying dial were used. Experience only determined the best angle of the driving-wheel axis with that of the grinding-wheel axis. It depended upon the type of work, kind of finish required, and the amount of stock to be removed. In the case of the tractor piston pin just described, an angle of 3 deg. was found to be the best.

Eleven center-type grinders had been replaced by three centerless machines producing the same number of pins and eliminating eight men. The production of pins per man had been increased 159 per eight hours. It might be mentioned that with centerless grinding on this type of work where precision was paramount, an absolutely dead-round diameter might be ground without any trace of chatter marks whatsoever, which on the contrary were very apparent in center grinding. After the final operation on the grinder the piston pin had, apparently to the eye, a perfect finish; but the lapping machine later installed showed up the imperfections of grinding by giving a mirror-like finish to the pin. The lapping machine was not used to rectify any inaccuracies in parallelism or roundness but merely to obtain an absolutely mirror finished surface. The production from one lapping machine was from six to eight pins per minute.

One of the most important advances in the perfection of the modern automobile power plant had been the accuracy obtained in the machining of pistons at low cost. The constant demand for

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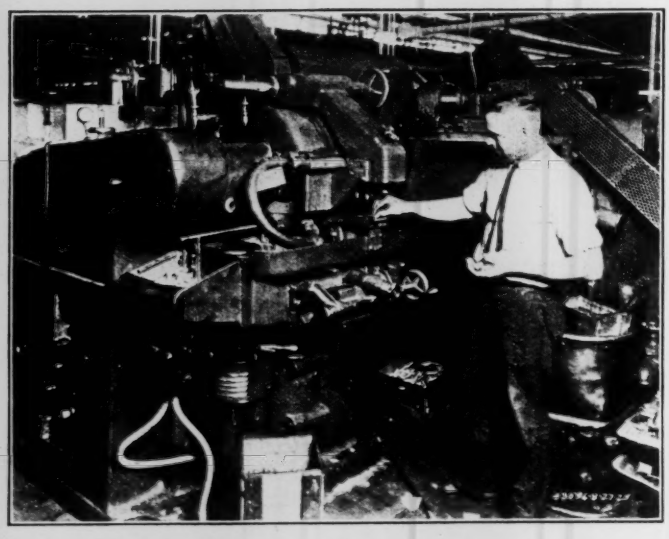
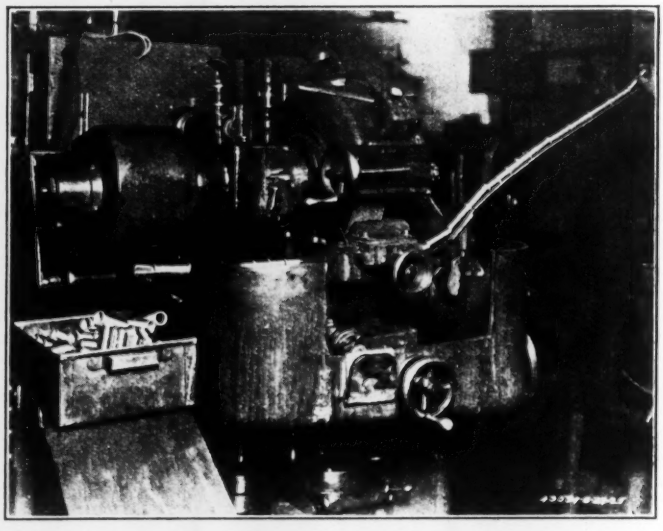
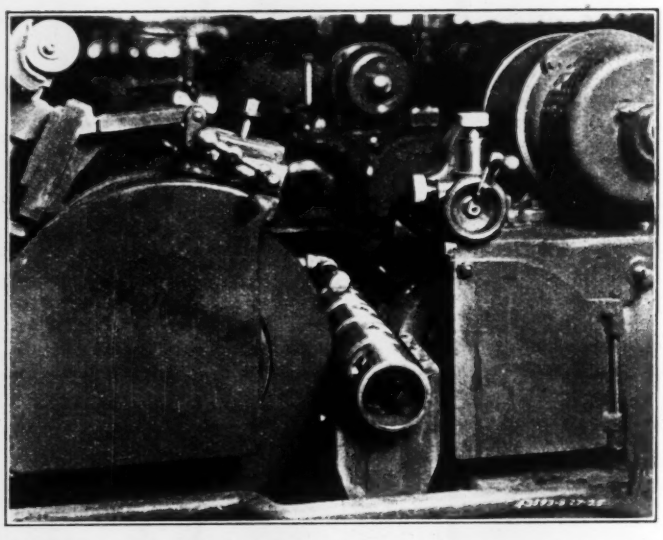


"IN FEED" METHOD OF CENTERLESS GRINDING

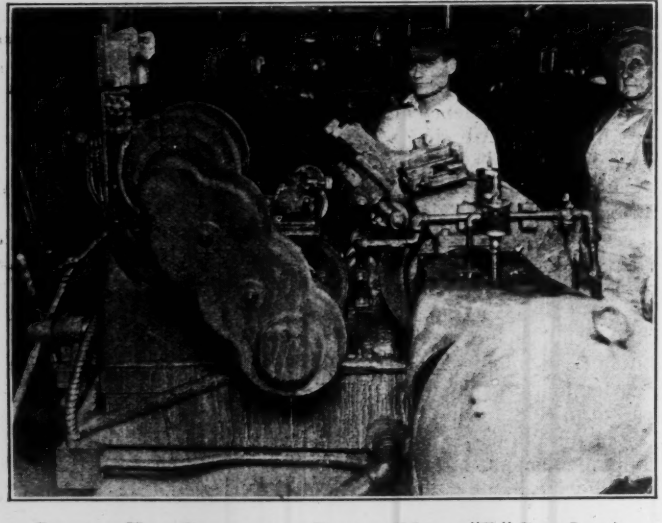


"THROUGH FEED" METHOD OF CENTERLESS GRINDING

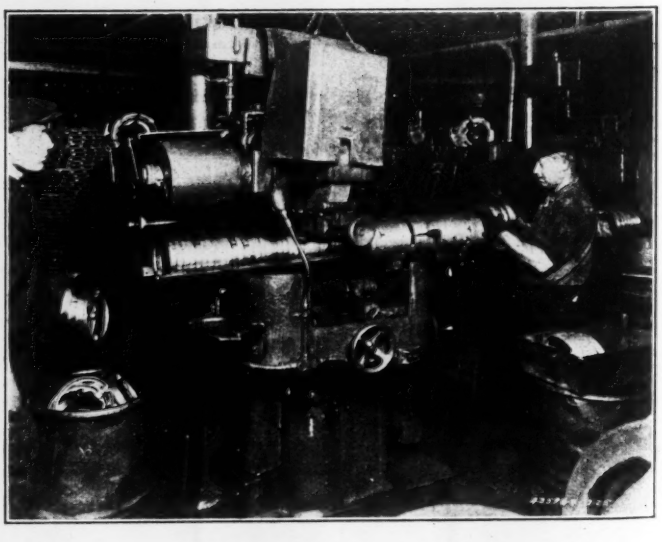
MODEL "T" PISTON PIN, HORIZONTAL-TYPE MACHINE



TRACTOR STEEL PISTON (0.045-IN. WALLS), HORIZONTAL MACHINE



GRINDING HEAT CLEARANCE IN PISTONS (MODEL "T," CAST IRON), HORIZONTAL-TYPE MACHINE



BRAKE SHOE, VERTICAL-TYPE MACHINE

smoothness and acceleration had forced the engineer to continually lighten reciprocating parts, to specify alloys and metals having greater tensile stress, and to demand closer limits in shop practice. This was especially true with the piston, and the centerless grinder had made it possible to machine lighter pistons with the greatest accuracy without the distorting effect of chucking—which was necessary in a turning machine—or of a cutting tool.

The light, close-grained gray-iron piston and the very much lighter steel piston used by the Ford Company, continued Mr. Seabury, because of their lightness had walls of thin section running only 0.045 in. in the case of the finished steel piston. The satisfactory chucking and machining on a lathe became more of a problem as the demand for lighter pistons increased. Even with special machines and special attachments the turning of pistons would soon be obsolete in his opinion because of the springing and distorting effect of the chucking and tool action on the thin-walled piston. The resistant force to turning set up by a cutting tool in machining a piston was more than one would estimate. This force had to be opposed by a strong chucking device. The result was a severe torsional strain on the thin-walled piston, which had proved to be one of the causes of elliptical or so-called "out of round" pistons.

It has been the Ford Company's experience that in the turning of pistons the intermittent striking and receding action of the tool in going over the piston-pin hole made it almost impossible to produce a round piston regardless of the care and precautions taken to use lighter cuts, to relieve a section around the piston-pin hole and in many other ways. It had not been unusual to find pistons having a wavy appearance the length of the piston after the finish-turning operation, regardless of the heat treatment or seasoning of the metal. The cause of this was attributed to the variation in the texture of the metal and the consequent springing action of the tool due to the variation in hardness commonly called "hard spots."

Centerless grinding greatly simplified the problem of finishing pistons accurately. In the case of the cast-iron piston the rough-turning operation was done first, after which the finishing operation was done by the centerless grinder. The amount of stock removed was 0.003 in. in the first operation through the centerless machine and 0.001 in. on the finish operation. Although the production ran about 50,000 pistons per day, the accuracy had been increased with the centerless machine so that the tolerance was now ± 0.0005 in. as compared with ± 0.002 in. when the finish operation had been a turning operation. The life of a grinding wheel on this job was about 50 days. The size of wheel was $20 \times 4 \times 9$ in. and it was dressed about four times in 24 hours.

In discussing the "through feed" method of centerless grinding, continued Mr. Seabury, he had dealt with work which had been done on the centerless grinder with the grinding wheel and regulating wheel mounted with their axes in a horizontal plane and opposite each other. The other type which he would discuss was the vertical opposed type having the grinding wheel mounted directly above the regulating wheel. The latter type had the advantage over the former for grinding narrow work of large diameter, as, for example, the Ford brake shoe, a casting about 1 in. wide and 8 in. in diameter, held to within $^1/_{64}$-in. limits in the foundry.

The 0.032 in. of stock to be removed was ground off in the vertical-type centerless grinder in two passes and the limits maintained were 0.002 in. for roundness. The cutting capacity was about five cubic inches per minute. This production had been obtained after persistent development of the machine as a whole, and especially the grinding wheel. It might be well to mention that at the start of the use of the centerless grinder on this part, 1500 shoes had been ground per wheel, but after considerable effort on the part of wheel manufacturers, the average shoes per wheel ran from 15,000 to 20,000, dressing the wheel only once at the start. The wheel now was used to within $^3/_4$ in. of the hole.

Formerly twelve center-type machines, with special heads and tail stocks had been used in grinding 1000 pins per machine for three shifts; now the same production was maintained with four centerless grinders running two shifts, eliminating 27 men. With the center-type machine the closest limit it had been possible to hold the brake shoes to was 0.010 in. for roundness; the present limits were 0.002 in. The breakage had been reduced from 600 to less than 100 per day. The speed of the grinding wheel on the first-operation machine was 1000 r.p.m. and on the second-operation machine, 1280 r.p.m. The size of grinding wheel used was $18 \times 4 \times 7^1/_2$ in. The vulcanite regulating wheel was $9^1/_2$ in. O.D. and 8 in. wide.

The relation of the work rest and the work to the grinding wheel and feed wheel was very important. For example, on the vertical type of machine the diamond was set at an angle of 7 deg. with the horizontal line going through the feed wheel and at 90 deg. with a line going through the center of the work, this line making an angle of 7 deg. with the vertical axis passing through the centers of the grinding wheel and feed wheel as shown in Fig. 1.

It was very important, wrote Mr. Seabury, to keep the grinding wheel sharp and free from gumming by using plenty of clean water with just enough soda to keep the machine from rusting. The compound recommended was 92 per cent water, 5 per cent plain soda and 3 per cent machine oil. The actual decrease in the cost of producing the brake shoe had been over 70 per cent with the centerless grinder.

In the development of the various types of centerless machines to meet the demands of the company's work, heavier bearing spindles than standard were used, also extra-large-diameter bearings having an abundant supply of clean lubricating oil. Greater accessibility of grinding and regulating wheels, extra-large motors for driving, and excessive capacity in the settling tank for cutting compound were necessary for continuous operation of the machines.

For finish-grinding the vertical type of machine seemed to have a slight advantage over the horizontal type. The regulating wheel should overlap the grinding wheel 2 in. on either side, causing the work to spin before and after reaching the grinding wheel and thereby preventing any nicking or flat spots on the finished surface. This applied to straight grinding or the "through feed" method only.

Shoulder grinding or the "in feed" method was done by having the grinding wheel oscillating. The regulating wheel for this class of work was of the same width as the grinding wheel. For automatic feed the vertical type gave better results both as regarded production and accuracy.

To obtain the best results with any grinding wheel a free cutting wheel was very essential. Analyzing the grinding wheel and observing it under a magnifying glass showed each small grain to be a cutting edge. If used too long without redressing it would cause the work to heat up and even draw the temper.

Centerless grinding was not confined to steel alone or to cast iron; it was also used to finish hard rubber, bakelite, fiber, vulcanite, carbon, soft rubber tubing, celluloid, bone, amber, porcelain, bronze, brass, aluminum, and square or hexagonal strip stock 20 ft. in length.

The problem of selecting the proper grinding wheel for various types of work was one requiring much study and experimental work, and the wheel manufacturers could supply valuable information along this line. The maximum efficiency of the grinding wheel depended upon the speed and grade of the wheel, its perfect balance, the grinding compound used, and the foundation of the machine in which it was employed.

A. L. DeLeeuw[1] in an oral discussion presented a novel theory on the relation between the control wheel, the cutting wheel, and the work. He did not believe that the work followed the slow or control wheel by the difference between friction of rest and friction of motion, but because it was easier for the fast-running wheel to chip off a small piece of metal than to give a large amount of momentum or energy to the piece of work. If a rod of wood were laid loosely on a table, one end projecting, and a sharp hatchet were brought slowly down upon the overhanging end of the rod, the rod would be knocked off the table. But if, on the other hand, the hatchet were brought down rapidly, the overhanging end of the rod would be cut off without disturbing the rest. Mr. DeLeeuw suggested that probably more or less the same thing happened when one tried to make round work out of oval work when supporting it on a V-block or angular support. In order to keep the work oval it would have to be shifted back and forth between the two wheels. Here again it was easier for the fast-running wheel to chip off a piece of metal than to give the necessary momentum or energy required to keep the piece of work moving from one wheel to the other.

James A. Hall[2] presented a written discussion in which he ques-

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[2] Associate Professor of Mechanical Engineering, Brown University, Providence, R. I. Mem. A.S.M.E.

tioned Mr. Peets' theory of centerless grinding based on the difference between the friction of rest and the friction of motion. He pointed out that the grinding wheel removed stock from the piece against which it ran, and this probably caused a different force relation from that which existed when two pieces rubbed or rolled on each other. James J. Guest in his book Grinding Machinery, wrote Professor Hall, stated that under good grinding conditions the normal pressure on the surface of the grinding wheel was about one-eighth of the tangential force. If this were considered a case of friction of motion, the coefficient of friction would be 8, or the force necessary to draw a piece of steel across a flat abrasive surface would be eight times that required to lift it. As this did not seem reasonable, Professor Hall felt that the theory of friction of rest versus friction of motion must be discarded.

The forces on a piece of work in a centerless grinder acted at the three points of contact, one with the grinding wheel, one with the work rest, and one with the regulating wheel, which tended to resist or control the rotation of the work. These forces were, of course, in equilibrium if the work rotated uniformly.

If these forces were drawn on a diagram of the machine and the limiting coefficient of friction of the work on the regulating wheel was taken as 1.0, which Professor Hall believed to be a reasonable figure, the ratio of the normal to the tangential force at the grinding wheel would become very large if slipping was not to occur at the regulating wheel.

In Professor Hall's opinion this large ratio of normal to tangential forces was one of the important considerations in comparing the theory of centerless grinding with grinding on centers. The reason for this greater pressure was probably because the wheel and the work rotated toward each other and the cutting went on in the approaching section of the arc of contact instead of the leaving section. This had been called "negative rotation."

This large normal pressure, Professor Hall believed, called for the use of somewhat harder wheels than for similar work when grinding on centers. In this his experience did not coincide with that of Mr. Peets.

The large normal pressure also accounted for the difficulty of maintaining the same diameter at the ends as at the center of work where the surface of contact was much greater, a difficulty which occurred particularly when heavy cuts were being taken. This difficulty also accounted for the necessity of more rigid spindle supports than when grinding on centers. It was probably also a factor in the difficulty of rapidly bringing out-of-round work within narrow limits of roundness.

Centerless grinding had made possible a remarkable increase in the production of many kinds of work and the process of determining the limits of its economical use was still going on. Professor Hall was not prepared to prophesy just what these limits might be, but wondered if this "negative-work rotation," with its accompanying large normal pressure in proportion to the tangential force, might not make finishing the primary function of this new type of grinder, and leave the removing of larger quantities of stock to the lathe or to grinding on centers.

In closing the discussion, Mr. Peets pointed out the difficulty of generalizing about the process of centerless grinding, as by so doing immediate conflict was discovered with some specific experience. He pointed out the importance of a mathematical analysis of the problems in centerless grinding.

Machine-Shop Sessions at New Haven

Excellent Exhibit, Well-Attended Sessions, Important Technical Committee Meetings, and Industrial-Plant Visits Make for Successful Meeting

THE New Haven Machine Tool Exhibition held from September 8 through 11, 1925, was an unqualified success. It combined an excellent exhibit of the latest developments in machine tools, four well-attended sessions on topics of vital importance to the machinery industries of the country, a series of round-table discussions on engineering problems in industry, and two important technical-committee meetings. There were also a series of visits to points of industrial and scenic interest in New Haven.

The auspices under which this convocation was held were highly favorable, as they included Yale University, the New Haven Chamber of Commerce, and the New Haven Section of the A.S.M.E. The Machine Shop Practice Division of the A.S.M.E. coöperated in the planning and conduct of the technical sessions.

The round-table discussions at the Lawn Club on Thursday evening proved to be a valuable part of the meeting. Here formality was laid aside and under active leaders at each table precision measurements, power-press work, production milling, and shop-training methods were discussed.

Two special research committees of the A.S.M.E. met during the meeting. On Wednesday morning, September 9, the Committee on Metal Springs discussed the preparation of a pamphlet of information regarding the scope of the investigation to be taken up. The Committee on Cutting and Forming of Metals met Thursday morning, September 10, and accepted the resignation of the chairman, Bryant H. Blood, who has now retired from active work and plans to travel extensively. Prof. James A. Hall, of Brown University, was selected as Mr. Blood's successor. Prof. O. W. Boston, of the University of Michigan, was appointed secretary of the Committee.

The opening session of the meeting was held on Wednesday afternoon, September 9, under the chairmanship of Frank O. Hoagland, Chairman of the Standing Committee on Professional Divisions of the A.S.M.E. A paper on The Theory and Practice of Centerless Grinding was presented by W. J. Peets. This paper appeared in the September issue of MECHANICAL ENGINEERING, and the discussion thereon is reported at length on another page of this issue.

O. B. Iles, President of the National Machine Tool Builders Association, was the principal speaker at the evening session on Wednesday, September 9, when he discussed The Future of the Machine-Tool Industry. Mr. Iles' address appears in abridged form in this issue of MECHANICAL ENGINEERING. The second subject was All-Metal Airplanes, the paper being prepared by William B. Stout, of the Stout Metal Airplane Company. Due to Mr. Stout's absence on the Pacific Coast, his paper was read and amplified by G. H. Hoppen, a close associate of Mr. Stout. Mr. Hoppen illustrated his presentation with lantern slides and moving pictures in a very interesting manner. The paper gave a description of metal airplanes and a recital of the engineering factors which must be solved in the design and construction of metal planes. Mr. Stout emphasized the fact that the use of metal—which has the advantages of permitting its properties to be determined with reasonable accuracy—enables the designer to calculate his elements more closely and provides a more economical structure than the use of wood and cloth structures. He said in part:

> We decided on metal as the only permanent type, and again, we decided on the use of one metal throughout. We were not in favor of steel spars with duralumin fittings or coverings, on account of the difference in the heat-expansion coefficients and their effect with such large areas and lengths involved. We wanted a wing of absolutely known qualities under any conditions, either of stress or temperature, and so decided to make the entire structure of duralumin.
>
> This metal is now so well known that it needs very little description except that it has the strength of structural steel and one-third the weight in its heat-treated form.
>
> After heat treating, the metal must be worked quickly before it sets, about half an hour being available for all the bending and formation processes. It is as non-corrosive as other metals, that is, with proper care and cleanliness there is no corrosion or rust. It is easy to work, and an extremely attractive metal for aircraft. We were able to obtain it from two different sources in the gages and forms we wanted.
>
> The basis of the wing design is of course the spar structure. The spar structure is in turn determined by the "chord" members. This chord shape consists of a semicircular rolled section riveted to a channel section. The

advantage of this semicircular shape is that the skin or outer surfacing can be riveted to the chord at any angle without any extra fittings. The channel section below gives side walls to which the vertical and diagonal spar members can be riveted.

Three cantilever spars are made up by this method, these being 32 in. deep at the center where the fuselage or body fastens on, tapering out to comparatively few inches at the tip. These spars are evenly placed inside the wings and intermediate former spars of like material are used for stiffeners to outline the wing contour in between. When these are in place, corrugated metal, rolled to fit the curve, is riveted to the spars top and bottom. In this way the skin becomes a part of the structure so that the actual surfacing of the plane is obtained for almost no weight, since it is performing the double function of surfacing and structure. The skin of this wing instead of being a liability to the structure as in cloth and wood types, is a definite structural asset.

The cantilever wing must be absolutely rigid, for which reason the only successful types in wood have been those covered with veneer, using the skin as part of the structure. This fact was found out in our original design and flying tests as far back as 1917.

It is for these reasons of torque rigidity that metal is the ideal material for all cantilever monoplane structures. In our own particular wing, once it is fabricated there is absolutely no flutter of any kind at any speed we have obtained, even in a dive, and no visible flexing of the wings even when fighting the most severe storms and bumpy weather conditions. This is due to the fact that the whole structure is homogeneous in both lateral and torque strength, and is more like a flattened-out tube arranged to take torque stresses, with spars inside to take definite load.

For shipping and production reasons, the wing tips are made detachable about half-way out from the body part. In metal, the fittings for such a detachable joint weigh but a few pounds, whereas in wood their weight and intricacy would involve 50 to 75 lb., and a very much increased expense and lessening of the factor of safety.

The wings run continuous across the top of the body or fuselage portion, and are fastened to it by six large bolts through gusset-plate fittings, fastening to the spar with a factor of safety double that of the wings.

The fuselage is built up on a jig arranged with a series of cross-bulkheads riveted up from duralumin channel sections. These sections are bent up in an ordinary tinner's brake and the entire set of bulkheads can be made by four men in two days' time, even with the simple equipment used.

The engine mount is also made of channel section, and is completely detachable from the body. The structure at the front end of the plane is increased in gage of metal for both strength and safety to the pilot.

As the structure goes back to the rear of the cabin, it is of very light material, and from there on to the tail, the load is carried entirely by the corrugated skin itself. The bulkheads, of course, are arranged in this portion for shear load. Diagonals are arranged in the cabin portion at the points where loads are carried from the wing structure down to the landing gear fittings so that stresses in this section are carried direct, and with very little expenditure of weight. This is ordinarily done with box sections rather than channel on account of the column loads involved.

The following are the general characteristics of this machine:

Span	58 ft. 4 in.
Length	46 ft. 2 in.
Height	12 ft. 2 in.
Weight, total	5849 lb.
Weight, empty	3717 lb.
Maximum speed	115 m.p.h.
Commercial speed	100 m.p.h.
Engine	400-hp. Liberty
Useful load	2132 lb.
Pay load with 4½ hr. fuel	1080 lb.
Climb first minute	500 ft.
Ceiling	15,000 ft.
Cubic capacity of cabin	250 cu. ft.

When the ship is completed, although it weighs less than 3800 lb., it is designed to carry 18 tons of sand on the wings as a test load without breaking, with a full factor of safety of 6; which means much more than the same factor of safety would in other indeterminate materials.

There are two things to be watched closely in the development of a commercial plane. First, it must be remembered that every pound carried is worth 20 cents per hour in the air. If one can save 100 lb., this means $20 per hr., and $200 per day loss on a 10-hour flying schedule. To add 100 lb. weight to a plane is to take $200 earning capacity per day off the ship—a feature entirely forgotten, seemingly, by many foreign commercial-airplane constructors.

Another item is of even greater importance, and that is, parasite resistance. For every pound you can save in head-on resistance of forward travel, you can add from 8 to 10 lb. of load to your plane, so that there are two methods of increasing the performance of a given plane with a given engine.

The final speaker of the evening was W. H. Rastall, Chief of the Industrial Machinery Division of the Bureau of Foreign and Domestic Commerce. In discussing the foreign trade outlook in the machine-tool field, Mr. Rastall presented statistics showing the growth of American machinery exports to various countries, and emphasized the need for immediately increasing the sales effort on behalf of American equipment in all foreign markets. He said in part:

A constructive sales policy would indicate that our manufacturers of the simpler types of metal-working machinery would be well advised if they gave more careful attention to the markets of Asia, and, in a general way, these remarks also apply to various South American countries. Mexico, Cuba, Argentina, Chile, and Peru are all absorbing a much larger volume of machine tools than in prewar years. Brazil is the only market that seems to us as falling off, and there are reasons to believe that this situation will correct itself as soon as the general business situation in that country improves.

The situation in Europe, however, is far more interesting and it is in these markets that our manufacturers of highly specialized production equipment find their best markets. In prewar years Germany ranked first, absorbing more American metal-working machinery than any other country, and this trade approximated $2,250,000 a year. Obviously, all of this had to be interrupted as soon as the war intervened, and even after the armistice no important sales could be made there because of import embargoes that were established by that government. Since the embargoes were lifted there has been a very rapid increase in the volume of business done, and throughout the latter part of 1924 and up to the present there has been a rapidly increasing flood of orders from German sources. Shipments to Germany rose from $77,000 in 1922 to $209,000 in 1923, to $545,000 in 1924, and for the first half of 1925 the total approximates $1,500,000, or, say, seven times the volume of business executed in the corresponding months of last year. When one considers the nature of the competition our manufacturers experience in the world's markets from German sources, this record is a marked testimonial to the superiority of American metal-working machinery.

In prewar years England ranked next to Germany as a buyer of American metal-working machinery and there has been a steady absorption of American machinery of these types in that market for a great many years. As compared with a prewar average of about $2,000,000 a year, England alone absorbed nearly $20,000,000 worth in 1916, and the volume shipped there continued very large throughout the war and the post-armistice boom. The low point in this trade was reached in 1922 when the shipments to the United Kingdom approximated $2,000,000, which in turn expanded to $2,400,000 in 1923, $3,100,000 in 1924; and experience in the first half of 1925 indicates a trade about 30 per cent in excess of the shipments made in the corresponding period last year. When it is remembered that British machine tools are perhaps the most serious competitors of American equipment in the world's markets, these returns also afford a very striking testimonial to the merits of the American product. For reasons that require little comment, the trade with Russia, Belgium, the Netherlands, and Sweden has shown some decrease, but considering the volume shipped to the United Kingdom, France, and Spain, and allowing for the conditions in Germany, Russia and elsewhere, it would seem that the markets of Europe have shown a very gratifying recovery, and considering the record of the past three years, it would appear that we have the adequate information needed to guide our foreign sales managers in establishing their sales policies for work in those markets. It seems that the time has come when conditions justify more careful sales work on behalf of American machinery in all of these markets, and the prospects for our trade depend not so much upon political and other conditions as upon the energy with which our products are represented there.

The third session of the meeting was held Thursday morning under the chairmanship of Ralph E. Flanders, past-president of the National Machine Tool Builders Association. Two papers were presented. Cylindrical Precision Lapping, by Paul M. Mueller, and High-Speed Cutting of Brass, by Luther D. Burlingame. Both of these papers appeared in the September issue of MECHANICAL ENGINEERING. The discussion thereon will be printed in the December issue.

An interesting and stimulating session on Education and Training for the Industries was held on Friday afternoon with Dugald C. Jackson, Professor of Electrical Engineering at Massachusetts Institute of Technology, in the chair. A paper by John P. Kottcamp, Assistant to the Vice-President, Johns-Manville Inc., of New York, brought out valuable and interesting discussion. This paper with the discussion will appear in the December issue of MECHANICAL ENGINEERING.

The exhibit of machine tools comprised the products of 110 manufacturers, and all of the available space in the Mason Laboratory, Sheffield Scientific School, at Yale University was occupied. The attendance throughout the exhibit was estimated at from fifteen to twenty thousand. A study of the exhibits clearly indicated that most machine tools of designs up to 1919 are now definitely obsolete. The progress made by tool manufacturers since they have been able to devote their time to development has placed a large amount of mechanical skill at the command of machine-tool users.

Plans are already under way for the exhibition in 1926, and tentative dates have been selected as September 7 through 10. The members of the Committee in charge of the Exhibition were H. R. Westcott, Chairman; Ernest Hartford, S. W. Dudley, W. W. Gaylord, J. D. Marsh, Nelson Brooks, K. F. Lees, Erik Oberg, Wm. Buxbaum, E. W. O'Brien, Charles Oppe, A. F. Breitenstein, George Holmes, G. P. Simpson, and Gustave Welter. The members of the A.S.M.E. responsible for the technical program are Erik Oberg, W. J. Peets, and John T. Faig.

MECHANICAL ENGINEERING

A Monthly Journal Containing a Review of Progress and Attainments in Mechanical Engineering and Related Fields, The Engineering Index (of current engineering literature), together with a Summary of the Activities, Papers and Proceedings of

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Gold from Mercury and Atomic Energy

A FEW weeks ago at the University of Cambridge the honorary degree of Doctor of Science was conferred on Prof. H. Nagaoka, of the Institute of Physical Research, University of Tokio, among other things, for his experiments on the transformation of mercury into gold. It is no exaggeration to state that only a few years ago Professor Nagaoka would have lost all standing in the scientific world had he dared merely to undertake such experiments.

From a paper which he presented to the International Astronomical Union, it would appear that extremely pure mercury was mixed with oil, put in a glass vessel, and subjected to electrical discharges of the order of 100,000 volts per cm. After ten to fifteen hours of exposure the oil had become carbonized and tiny specks of gold could be found everywhere in the mixture of mercury and carbon. Minute gold particles even penetrated the glass, which had become a ruby color.

Mercury consists of a nucleus and 80 electrons traveling around it, while gold contains only 79 electrons; so that by knocking out one electron of mercury, gold might be produced under the assumption that the remaining 79 electrons would be compelled to rearrange themselves in the manner in which they are arranged in gold, i.e., assuming that the electron orbital pattern is definitely determined by the number of electrons about the nucleus. All indications are to the effect that this is so.

These experiments give support to the experiments previously announced by Professor Miethe in Germany. The process as yet does not have any economic importance as only the production of extremely small particles of gold at best has been achieved, and this has been done at a cost, with our present sources of power, many hundreds of times that of natural gold.

To the mechanical engineer, however, this may prove to be of enormous interest. An incentive is given thereby to more intensive work on the constitution of the atom and the possibilities lying in an artificial breakdown of atom structure. While the work of Miethe and Nagaoka is directed toward the breakdown of more complex atoms into less complex atoms, such as mercury into gold, the information thus gathered may and probably will sooner or later give an inkling as to the possibilities of the inverse process, viz., transformation of simple atoms into complex ones, possibly with the liberation of a large amount of energy.

It is easily conceivable that if to transfer mercury into gold an amount of energy many times the value of the gold has to be employed, by transforming gold into mercury a similar amount of energy would be liberated; and the day may not be far off when, for all we know, bars of gold instead of tanks of oil or carloads of coal may form the "fuel" of our prime movers.

As a matter of fact, however, it is not at all necessary to limit our future sources of energy to the comparatively rare metal gold. If energy can be released by the transformation of atoms of simple structure into more complex atoms, it should not be forgotten that the simplest atom of them all, hydrogen, constitutes a substantial share of the weight of our planet and is everywhere available in combination with oxygen as water, and can be easily recovered therefrom.

There is a growing feeling among mechanical engineers to the effect that our prime movers—boilers, turbines, engines—have each within their field reached the maximum efficiency of which the present types are capable, and that progress can be achieved more profitably by some revolutionary development rather than by the way of gradual improvement. How far off are the developments dimly promised by such work as that of Professor Nagaoka, no one can tell. The fact remains, however, that physicists apparently consider this line of research worthy of serious consideration, and mechanical engineers would do well to keep an eye on what the physicists are doing in this direction.

The Novelty of Standardization

THAT numerous present-day problems were anticipated and partially solved by early leaders of thought and action is well known. It is not unexpected, therefore, to find that the standardization of machine-tool elements, now a subject of much discussion, was advocated in a forceful way by William Lodge, a pioneer in machine-tool construction, a business statesman, and a founder of the National Machine Tool Builders Association. In an address in 1903 before the N.M.T.B.A., quoted by August H. Tuechter in a recent issue of *Army Ordnance*, Mr. Lodge dealt with this subject as follows:

> It would be to the interest of this association if it could possibly be brought about that some standard be adopted. Why may not all manufacturers of machine tools have the same size of general bearings, the same size of nose for the lathe spindles for the different sizes of lathes, so that our customers may find it easy to transfer face plates, chucks, and tools from one machine to another, irrespective of who may have been the manufacturer? If this matter can be brought about we shall be the first country to have adopted a uniformity. This would be a most important matter, not only to ourselves, but to the men to whom we sell.

At the time of the presentation of this address high-speed steel had been discovered four years and a thorough revolution in the design of machine tools was about to take place. The situation now facing the machine-tool manufacturer is much the same as it was in 1903, although a number of minor factors are replacing the major factor involved in the adoption of a new cutting material. At the present time designs of metal-cutting tools have advanced to the point where machine tools regarded as highly efficient for production during the war peak must now be regarded as obsolete. Despite Mr. Lodge's earnest plea for standardization, it has only been in the last two years that definite projects have been taken up looking to the standardization of machine-tool elements. Progress is being made, but the standard for only one machine element is nearing completion.

Today machine-tool builders tend to regard the standardization of machine-tool elements as an activity that will eliminate the originality of the designer and hinder the rapid development of metal-cutting tools. The users, however, need tools with standardized tool-holding and work-holding elements. Fortunately the procedure of the American Engineering Standards Committee provides for adequate representation of the users of machine tools on standardizing committees and their influence should be helpful and stimulating in securing concerted action in these projects.

The novelty of the present situation arises from the fact that twenty-two years ago, Mr. Lodge, a far-sighted leader of the counsels of the N.M.T.B.A. stated the users' case so clearly and emphatically.

"Railroad Men"

TO THE individual whose contact with railroads has been confined to an occasional difference with a lattice-framed abrupt ticket seller or to an argument with a pompous, rule-bound train conductor, the recent A.S.M.E. meeting at Altoona revealed a new type of railroad man, the professional engineer, a man who has dedicated his talents to railroading, who glories in his calling, and who sacrifices many of the comforts of life and the possibilities of professional advancement that he may contribute to the technical development of the great transportation systems that make American civilization possible. Railroading is a game. Its spirit crowds the veins of these quiet, modest, and efficient men. It baffles analysis that a man bred a "railroader" is permeated with a mysterious urge that keeps him a "railroader" for life. Perhaps it's the difficulty of the game, or its ever-changing novelty, or the bit of danger that comes in governing great mechanical forces that grips these men and holds them for a life career. But whatever the tie, it develops men of strength, courage, and squareness, whose pride in their calling provides an unwritten code of high ethics and a devotion to duty and the service equaled by no other group of professional men. They carry on in their unswerving attack on their daily problems without the encouragement of public appreciation of what they are doing to advance the well-being and industrial growth of the nation. But they carry on with a dignity and modesty befitting the tradition of generations of engineers whose achievements have made railroads the basic economic force they now are. The apparently unlimited demands that these men calmly meet and fulfil under the difficulties of complicated organization and minute governmental regulation of technical detail, react in them to a secret satisfaction that is shown only by the unwilling admission that they are "railroad men."

Patent Office Procedure

EARLY in September, 1924, the Honorable Hubert Work, Secretary of the Interior, requested several patent-law associations, the Chamber of Commerce of the United States, the National Manufacturers' Association, and American Engineering Council to nominate representatives to serve on a committee to make a study of Patent Office procedure. Each organization complied and the Committee began to function in October, 1924, with the following personnel:

HON. THOMAS EWING, *Chairman*, of New York City, formerly Commissioner of Patents; nominated by the New York Patent Law Association;

JO. BAILEY BROWN, of Pittsburgh, Pa., one of the patent advisers representing the United States at the Paris Peace Conference; nominated by the Pittsburgh Patent Law Association;

A. J. BROSSEAU, of New York City, a prominent automobile manufacturer; nominated by the Chamber of Commerce of the United States;

COL. HARRY FREASE, of Canton, Ohio, formerly president of the Cleveland Patent Law Association, and nominated by that Association;

HENRY M. HUXLEY, of Chicago, Ill., Secretary of the Chicago Patent Law Association, and nominated by that Association;

EUGENE G. MASON, of Washington, D. C., formerly Secretary of the Patent Section of the American Bar Association, and nominated by the Hon. Charles E. Hughes, President of that association;

GEORGE A. PREVOST, of Washington, D. C., Vice-President of the American Patent Law Association, the national association of patent lawyers, and nominated by that association;

EDWIN J. PRINDLE, of New York City, formerly Secretary of the Patent Committee of the National Research Council; nominated by the National Manufacturers Association;

MILTON TIBBETTS, of Detroit, Mich., Assistant Secretary of the Packard Motor Car Company, formerly President of the Michigan Patent Law Association, and nominated by that association;

L. W. WALLACE, of Washington, D. C., Executive Secretary of American Engineering Council, and nominated by that council.

HENRY N. PAUL, JR., of Philadelphia, Pa., patent attorney; nominated by the Philadelphia Patent Law Association.

When the Patent Office was transferred to the Department of Commerce, Secretary Hoover requested the Committee to continue its work. However, he added to the Committee Messrs. Wallace Clark and W. H. Leffingwell, who were nominated by The American Society of Mechanical Engineers.

At the first meeting, the Secretary of the Interior addressed the Committee as follows:

Recognizing the great value of the Patent Office in the development of the industrial life of the United States, I am especially desirous of doing anything possible to facilitate methods of procedure and to expedite the business of that office.

At the present time the office is far behind in its work. This is due in a large measure to the tremendous increase in applications for patents, which began shortly after the close of the war and has continued ever since. Although the force of employees has been increased on two occasions by Congress, the Patent Office is still in arrears. An application for a patent filed today will not be acted on until seven or eight months later.

The Commissioner of Patents has experienced great difficulty in his effort to bring the work to a current basis and for this reason, with his approval, I decided to ask for outside assistance. In all institutions where methods of procedure have gradually developed, the general tendency is that such procedure becomes more complicated. This is particularly true where both technical and legal questions are involved in the same cases.

A very useful purpose may be served by this committee of prominent patent attorneys and representatives of industrial organizations, including the engineering profession, in making a thorough review of the Patent Office with a view of simplifying the methods of procedure and expediting the handling of applications for patents.

Unfortunately, the Interior Department has not available any funds for the purpose of paying the expenses of the Committee and your work, therefore, will be a voluntary contribution, rendered in the spirit of public service. The Commissioner of Patents will give you every possible aid in your study of various functions of the Patent Office and assist you in every possible way in your survey of its affairs. Office quarters, stenographers, and all necessary clerical assistance will be supplied.

I shall look forward with the greatest interest for your opinions and at the conclusion of your inquiry, I shall appreciate receiving a report from you with such recommendations as you may deem necessary to improve conditions and shall submit it to the President for his information.

The Secretary of the Interior gave further instructions to the Committee in a letter of October 11, 1924, in which he said in part:

Your study of the Patent Office, therefore, may properly cover not only an inspection of the methods and review of procedure now in effect, but consideration may profitably be given to the amendment of the patent laws wherever the need is indicated, with due regard particularly to the full protection of the interests of the Government and the inventor.

Experience has shown the necessity of the periodical weighing of operating methods in any large organization. Particularly is this true in a unit of the Government which has had a long and progressive growth, as has the Patent Office. Good organization is constant reorganization from within, but gradual growth frequently produces duplication of methods in any enterprise. Old precedents now having the force of law were once new precedents and as such, perhaps, displaced others which have been followed implicitly for a long term of years. But precedents, no matter how well founded at the time of their adoption, may be no longer applicable under the changing conditions of our industrial life.

Accordingly, I would indicate efforts especially directed to the simplification of office methods; abbreviating formality in procedure, reducing the number of forms, and the amending of the patent laws where they present an obstacle to progress. No phase of the work is so small as to be unimportant; nor, on the other hand, should any subject be too large and far-reaching to deter you from approaching it from a practical viewpoint if you believe it to be fundamentally wrong.

After receiving its instructions the Committee seriously undertook the important task assigned to it. It has held nine sessions of two days each. The several sub-committees have spent many days studying various phases of the work. Hundreds of suggestions have been reviewed by a sub-committee and acted upon by the Committee. A number of aspects have been considered and acted upon by patent-law associations and the findings debated by the Committee. The report, therefore, will represent the thought and work of many minds.

The Committee contemplates submitting its report to the Secretary of Commerce early in December. The report will contain over one hundred recommendations. They will cover such matters as a fundamental change in the organization of the Patent Office; an exposition of the intolerable housing situation; the grave need for a modern building and standard equipment; obtaining, training, and maintaining an adequate technical staff; need for expediting and improving upon the classification of patents, a material revamping of the present methods of executing the work of the office; fundamental changes in the manner in which interference and amended cases and appeals are treated; shortening of period of time allowed for responses to office actions; and revision of existing statutes and rules in some directions. All of which is being recommended for the purpose of enabling the Patent Office to do a better quality of work and to shorten the time required to prosecute an application for a patent.

The beneficial results contemplated can be realized only through

all concerned becoming acquainted with the objectives and throwing the weight of their influence in support thereof. The directions in which support will be needed will be obvious when the detailed findings of the Committee are made known. A subsequent communication will contain some of the major findings.

L. W. WALLACE.[1]

The Pulitzer Race of 1925

THE Pulitzer Trophy race and the nine events that generally accompany it constitute the great aeronautical event of the year. The races of 1925 run at Mitchel Field, L. I., from October 8 to 13, provided thrills for the spectators, as well as many suggestions for improvement in plane and motor design.

In the very first event of the race, a terrible accident marred the proceedings, C. D. Chamberlin, a skilled pilot of Hasbrouck Heights, N. J., had remodeled a biplane built in 1918 and obtained an uncertain structure; this crashed a few minutes after taking off, killing the passenger carried and injuring the pilot.

What more typical of the changing character of American aviation than the appearance of a Ford Airliner, to be used with nine others in a winter express service to Florida! Fokker exhibited his latest transport plane, a three-engined passenger monoplane of 10-seat capacity equipped with 200-hp. Wright-J motors. A controversy has long existed between advocates of the single-engined and the multiple-engined airplane. The new Fokker was a powerful argument in favor of the latter type. Its three motors were placed close together, the two outer ones swinging their propellers within the circle of the forward propeller disk. As a result, when an outboard engine fails, the unbalanced couple that results is relatively slight. Exhibition trials showed that continuous flight on any two of the three motors was possible.

The meet brought into prominence a series of light transport planes of considerable interest, such as the Wright Bellanca, the Sikorsky S-31-A, the Aerial Service Co.'s Mercury Junior, and one or two others. The granting of contracts to private air-mail operators has stimulated the production of such planes of about 200 hp., capable of carrying a mail load of 500 lb. and of serving the feeder lines of the transcontinental airway.

Efficiency prizes for planes of less than 200 hp. were awarded on the basis of multiplying the average speed in the race by passenger and baggage load and dividing it by the horsepower. In 1924 the value thus obtained was 430 points. This year the Wright-Bellanca, with a 200-hp. Wright air-cooled motor, achieved 602 points at the comfortable speed of 112 m.p.h. (including time lost turning pylons), carrying five passengers and baggage equivalent to 200 lb. The Wright-Bellanca showed further that the ingenuity of designers is not as yet exhausted. The struts used to brace the monoplane wing were in themselves lifting elements and the landing gear had two sturdy legs built on the cantilver principle without external bracing to introduce air resistance.

The speed of the Pulitzer racers has steadily increased in successive years. In 1920 it was 156.5 miles per hour; in 1921, 176.7; in 1922, 205.8; and in 1923, 243.68. In 1924, when no new planes were built, the figure fell to 215.72; but this year, with two years' development, great progress was expected. Accordingly the new figure of 248.99 established by Lieut. Cyrus Bettis of the Army, was a disappointment to both air services. The Curtiss Company had made no very striking changes in the design of the plane but it had built to Government order twelve motors, termed "V-1400," which were rated at 619 hp. for a weight of only 660 lb. or 1.1 lb. per hp., at 2500 r.p.m. On the block these figures had been realized with one motor. But in the air the motors ran at approximately 2300 r.p.m. instead of the 2500 expected. In high-speed plane design, barring radical departures from present practice, it would therefore seem as if further increase in speed will be painfully gained mile by mile. However, this should not be taken as invalidating the usefulness of the Pulitzer races. The Army Air Service pursuit planes gave over 161 miles per hour with full military equipment. They are the direct derivatives of the Pulitzer racers, with only minor changes in design such as increased fuselage space and larger wings.

Another controversy, water cooling versus air cooling, appears to be in process of solution. The air-cooled motor dispenses with the heavy radiator and cooling water, but in heavy powers and large cylinder sizes it has not reached the development of the water-cooled type. Up to 200 hp., however, the air-cooled engine seems likely to displace the water-cooled one. This was shown by the preponderance of air-cooled-engined planes in the new types, Fokker, Wright-Bellanca, Sikorsky, etc.

It was interesting to see the Caminez engine in which the pistons work directly through roller bearings on a large cam mounted on the drive shaft. The four-cycle engine of this type allows each piston to make four strokes for one revolution of the engine. In this way high piston speeds are maintained without excessive propeller speeds, and valve timing gears are eliminated, the result being a compact and simple engine.

Perhaps the wish is father to the thought, but it would appear that the outstanding feature of the meet was a shift of interest, at least on the part of technicians and manufacturers, from the racers to the commercially promising transport planes.

Cleveland Safety Congress

UNDER the auspices of the National Safety Council, the Fourteenth Annual Safety Congress was held in Cleveland, September 28 to October 2. During the course of the Congress over 250 speakers, many of them internationally known, addressed the meetings. The presiding officer was President C. B. Auel, of Pittsburgh, also a member of the A.S.M.E. and active on its Committee on Safety.

Some of the outstanding addresses included those made by Rear Admiral William S. Sims on Safety in the Nation, by Hon. John Barton Payne, Chairman of the Central Committee of the American National Red Cross, and by Carl R. Gray, President of the Union Pacific Railway System.

Charles B. Scott, in charge of Industrial Relations for the Insull Interests of Chicago, was elected president for the ensuing year.

A number of sessions interesting to members of the A.S.M.E. were held, these including discussions of safety problems dealing with cement, rubber, forging, refrigeration, marines, metals, paper and pulp, petroleum, public utilities, electric railway, textiles, and woodworking.

An interesting phase of the Congress was an exhibition of safety appliances and equipment, a number of the exhibits having to do with products requiring mechanical engineering in their manufacture.

At the sessions of the American Society of Safety Engineers, the engineering section of the Safety Council—and which is affiliated with the A.S.M.E., Dr. A. D. Risteen, Mem. A.S.M.E., presided. The reports of a number of committees were made at these sessions, and there was considerable discussion on a paper presented by Dr. R. H. Lansburgh, Secretary of the Pennsylvania Department of Labor and Industry, and dealing with Safety Codes—Their Adoption and Enforcement. It was very evident that there is need of much closer coöperation by the engineers of each state with those charged with making and enforcing laws having to do with safety codes. The woodworking section contributed a number of papers of interest to members of the A.S.M.E., including four respectively entitled, The Elimination of Power-Transmission Hazards In Older Mills, The Evolution of Safety and Woodworking Machinery, Safety Guards for Mechanical Skidders, and Exhaust Systems in Woodworking Shops. The metal section was also responsible for several interesting papers, including one dealing with the transportation of steel products safely, and another with the handling of molten metal in the foundry.

In his presidential address, Mr. Auel, after pointing out what the National Safety Council had accomplished in the fourteen years of its existence and the manner in which it functioned, stated that the Council was organized originally to cover the field of industrial safety; but the problem was not the simple one of guarding equipment and installing safety systems. Developments showed that it was not the industries alone which needed to be taught accident prevention, but the country generally.

A number of motion-picture films showing safety methods used in industry plants were shown at the Congress and many of these will be available for use at the meetings of A.S.M.E. Local Sections.

[1] Executive Secretary, American Engineering Council, Washington, D. C. Mem. A.S.M.E.

Distinctive Railroad Meeting at Altoona

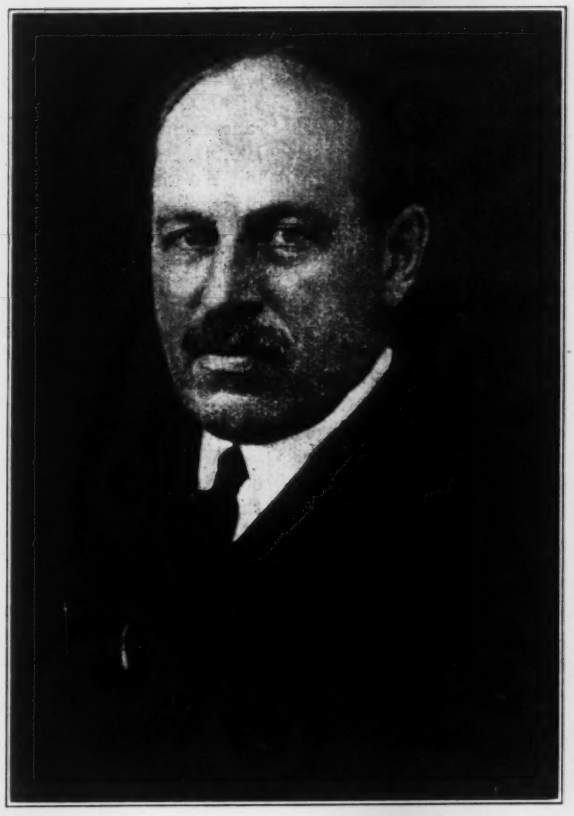
Three-Day Convention of Engineers Marked by Discussion of American Transportation Problems—Addresses Made on the Occasion of Conferring Honorary Membership on Gen. W. W. Atterbury, President of the Pennsylvania Railroad System

THE Altoona Meeting of The American Society of Mechanical Engineers was a splendid occasion. The three days of October 5, 6, and 7 were taken up in a program that included the dignified ceremony of conferring Honorary Membership in the Society upon General William Wallace Atterbury, the serious discussion of American transportation problems, and a series of delightfulexcursions and entertainments that attracted a distinguished group of railroad executives and engineers. Altoona is a delightful spot to hold a meeting, surrounded as it is by the beautiful scenery of the Alleghany Mountains; but it is an especially desirable place to hold a railroad meeting, as it is a distinctively railroad town and its atmosphere conducive to that type of professional fellowship which is important in a meeting of that kind.

The Pennsylvania Railroad System and the Central Pennsylvania Section of the A.S.M.E. acted as hosts for the meeting, and the meticulous care with which the events were planned and conducted redounded greatly to their credit. An especially interesting program was provided for the ladies, and those who attended were royally entertained.

The attendance at the meeting was over two hundred members, guests, and ladies, but this does not include over one hundred members of the A.S.M.E. Student Branches at Pennsylvania State College and Bucknell University who ascended and descended to Altoona in "flivvers" and trucks to hear the words of wisdom that abounded at the sessions, to see the Altoona Works, and to absorb the professional atmosphere of an A.S.M.E. meeting.

Monday, October 5, was given over to an all-day meeting of the Council, the first of the fiscal year, at which the budget was approved and other routine business transacted. This meeting was attended by President-elect W. L. Abbott and Councilors-elect W. T. Magruder, Charles Gorton, and William Elmer. The *A.S.M.E. News* of October 7 contained a report of the important transactions at the meeting.



W. W. Atterbury

On Monday evening Honorary Membership was conferred upon General William Wallace Atterbury, who took office as president of the Pennsylvania Railroad System on October 1. The ceremony took place at the Hotel Penn-Alto following a dinner to the railroad officials and members of the A.S.M.E. Council at the historic Logan House. Previous to the ceremony, which was presided over by Dr. William F. Durand, President of the Society, there was a short program of excellent music by the choir of the Altoona Works of the Pennsylvania Railroad System, and an address of hearty welcome by Elisha Lee, Vice-President in Charge of Operation of the Pennsylvania Railroad System.

Mr. Lee pointed to the pride which railroad men have in their calling, and acknowledged the debt of railroads and railroad managers to the mechanical engineering profession for their valuable contribution to the main purpose of the railroads, which is the rendering of transportation service. He called attention to the fact that the A.S.M.E. had met in Altoona in 1881, and in bidding the Society welcome on the occasion of its return to this center of mechanical activity of the System, he emphasized the great strides in railroad methods that had taken place since the early meeting.

Dr. Durand expressed gracious appreciation for the cordial welcome and before proceeding to the main business of the evening acknowledged the presence of Honorary Members Charles M. Schwab, Worcester R. Warner, and Ambrose Swasey. He also spoke of George M. Bond, who with Mr. Swasey was present at the meeting in 1881. He then called on Charles Day, of Philadelphia, who presented General Atterbury to the President for the bestowal of the certificate of Honorary Membership. Mr. Day's words and General Atterbury's response are given in full below, for they have much significance.

Mr. Day spoke as follows:

"When The American Society of Mechanical Engineers was organized in 1880, provision was made that the Society's highest distinction, Honorary Membership, should be conferred only in recognition of acknowledged professional eminence.

"No one who reads such names as Orville Wright Goethals, Schwab, Westinghouse, Parsons, Vauclain, Rateau, Warner, Swasey, Edison, and indeed the entire list of those who have had this honor conferred upon them, can fail to realize that the original purpose of our Honorary Membership has been zealously guarded, and that in every instance only engineering service of the most distinguished order and of undeniable value to mankind has been so rewarded.

"The American Society of Mechanical Engineers, after careful deliberation and the formality leading to such decision, will admit to this small group of Honorary Members tonight another engineer, William Wallace Atterbury. It is my rare privilege, Mr. President, to outline briefly the reasons which prompted this action.

"Mr. Atterbury entered the service of the Pennsylvania Railroad on October 11, 1886, having been graduated from Yale University that same year. Since then he has been actively and continuously identified with that organization except for the period when our country participated in the Great War. It is appropriate that we should confer this honor on him in Altoona, since he started as apprentice in the then comparatively modest shops of the Pennsylvania Railroad located here. Successively he has occupied the positions of Assistant Road Foreman of Engines; Assistant Engineer of Motive Power on the Northwestern System; Master Mechanic at Fort Wayne; Superintendent of Motive Power, Lines East; General Superintendent of Motive Power; General Manager of Lines, East; 5th Vice-President in Charge of Transportation; Vice-President in charge of Operation; and now President of the Pennsylvania Railroad—39 years of continuous service in the interest of this, one of the greatest railroad systems of the world. Such a record in itself speaks for his industry, capacity, and ability through those early years when, step by step, he was ever fitting himself for posts of greater responsibility. Obviously, engineering talent of a very high order made possible General Atterbury's enviable career in the field of American steam-railroad transportation, but in addition it was because he possessed great ability as an administrator and

keen business acumen that his outstanding achievements in more recent years have been possible.

"Among railroad officials, not only in the United States but throughout the world, his work for years had attracted wide attention and had prompted the highest commendation; so in 1917 it was but natural that the searching inquiries of the War Department for the railroad executive best qualified to handle our railroad problems in France should point to the selection of William Wallace Atterbury for that tremendously important post. He sailed for Europe in August, 1917, as a civilian, desiring to study the situation in all of its aspects before accepting a commission, and on September 14 became Director General of Transportation of the American Expeditionary Forces. On the fifth of October that same year, following the recommendation of General Pershing, he was commissioned Brigadier-General of the United States Army.

"Noteworthy as had been Mr. Atterbury's contributions in the field of steam-railroad engineering and operation, we must nevertheless follow his career in France before we can appraise to the full his great capacity and ability and the larger service he was privileged to render his country. Consequently I will attempt, in the few moments remaining at my disposal, to carry you back to those months of unprecedented trial and strain, recalling to your memory particularly the transportation task which had its beginning when our ships dropped anchor in the French ports and continued until their burdens, whether men or munitions, were delivered to the American fighting fronts.

"The Ocean Transport Service was created to unload and "turn around" the ships. The Transportation Corps was charged with the operation, maintenance, and construction of all railroads turned over to the American forces by the French, and the supervision of all movements of American troops and supplies over French lines. As our regular army had not had experience in railroad operations, it required but a few months to emphasize the absolute necessity of calling upon men who had made such problems their life work.

"When General Atterbury took hold of the Transportation Corps he realized at once that it would be impossible to foresee with certainty the scope and volume of the work involved or the best manner of handling it, for he was confronted with an overseas military program in comparison with which all previous expeditions dwindled into insignificance.

"The first and paramount cause for constant readjustments of plans and procedures was the rapidity with which America's participation increased. New construction became of first importance, and much of it was taken over by the Army Engineer Corps. All other great departments embracing the Service of the Rear were being enlarged and modified in their scope and relationship to each other. There were also the problems growing out of the interrelationship and division of responsibility between army and civilian personnel. Finally there was the necessity of working in the most intimate relations with the French authorities. Consequently, from the moment General Atterbury accepted his commission until the armistice was signed, he was constantly concerned with questions of organization presenting a complexity and subtlety, coupled with a measure of importance, which can be understood and appreciated only by those who were themselves intimately identified with them.

"Among the most important changes affecting his work was the coördination brought about between the Transportation Corps and Ocean Transport Service through placing them both under the control of the Director General of Transportation. Within six months after General Atterbury's arrival in France these two departments embraced, under his supervision, over 1600 officers and 60,000 men.

"The briefest reference to the work itself should start at the ports, which presented a stupendous task of expanding French facilities and providing entirely new works. To equip the ports of Brest, St. Nazaire, Bordeaux, and Marseilles—to mention only the principal ones—for the cargo movement in prospect was a vast undertaking necessitating the use of endless shiploads of materials and machinery from the United States, embracing docks, handling equipment of every description, shops, and terminal facilities. Visualize, too, the reconstruction of hundreds of miles of main-line railroad, planned and the construction supervised by the Director General's organization, the provision of much new equipment, including over 1500 American locomotives and 20,000 freight cars, the planning and designing of the necessary shops and the coördination of the entire work with the great army storage areas, munition dumps, military encampments, aviation fields, hospital centers, etc. And finally, when you have grasped all this, let me remind you we are considering only the preparatory service that General Atterbury had to dispose of before the job of troop and cargo movement could go forward on a large scale.

"I have referred already to the rapidity with which the picture changed. In October, 1917, we had only 100,000 men in France and the enemy submarines were sinking a million tons of shipping every thirty days; consequently a troop movement at the rate of 75,000 to 100,000 men per month was an expectation of but a small and tremendously optimistic minority, and then only as a possibility many months hence. Nevertheless such miracles, seemingly, were wrought at home and on the seas that during the early fall of 1918 when the great German offensive was under way, General Atterbury was called upon to handle troops at the rate of 300,000 per month, and to include in his operations new American battlefronts never contemplated in the original plan. I will leave to your imagination the obligation that this great troop movement imposed for the continuous forwarding of increasing supplies of food, munitions, clothing and equipment. Just prior to the armistice almost 1,000,000 tons of supplies were being handled per month, and arrangements were being rushed for three times this amount, or 100,000 tons per day—a stupendous task under the best conditions; but the job had to be done under the pressure of war, and furthermore in large part through existing French railroad organizations.

"Harassing and at times almost unsolvable difficulties were of daily occurrence, a condition inevitable in such a stupendous organization created almost over night from army and civilian personnel. But, throughout his services for the Army, General Atterbury steered his course with an eye only to the single problem of winning the war. His own fortunes suffered violent changes. He was not saved from the experiences of other civilian officers in this respect, but he hung on to the job and fought it through in spite of every kind of difficulty and opposition.

"After the resumption on an unprecedented scale of military operations early in 1918, it was a constant race between three agencies: those who were moving troops and supplies to our American ports of embarkation, the Shipping Board which supplied vessels for the transatlantic haul, and those great departments in France which, under General Atterbury's leadership, unloaded the ships and forwarded the men and cargoes. Responsibility for delay was placed at different times on each of these departments, and finally the controversy between 'ships' and 'port facilities' became so acute that Mr. Hurley's and General Atterbury's departments became veritable storm centers. I mention these details that you may better understand the circumstances under which the work was prosecuted. So vital were the issues, that our Commander-in-Chief, notwithstanding the gravity of the situation at the front, made a searching personal inquiry visiting every major port, investigating favorable and unfavorable report, only to serve an ultimatum that General Atterbury's splendid engineering and administration achievements deserved nothing short of the fullest support that he could accord, and that henceforth he would look to him and him alone in connection with the ports and the rail transport.

"If General Atterbury had not been gifted with rare ability and capacity and if his experience in the fields of engineering, construction, and operation had been less inclusive, and finally if he had not been a life-long student of human relations, he could never have filled a post of such immense importance with such outstanding success throughout our participation in the war.

"He returned home early in 1919, decorated by his own country, by Great Britain, France, Belgium, and Servia. He resumed his work for the Pennsylvania Railroad, tackling with courage new and difficult problems growing out of the war—a period of unprecedented confusion, misunderstanding, and difficulty following the return of the railroads to their owners. His has been a sustained interest in the continued development of American transportation to which his life has been devoted. His services as President of the American Railway Association are known to most of you.

"Mr. President, I could say a great deal more about the work of our distinguished fellow-member, but his work and service need no spokesman. They are recognized as constituting engineering

achievement which has proved of outstanding service to mankind.

"On behalf of the Council of The American Society of Mechanical Engineers, it gives me great pleasure to present to you, that you may confer on him Honorary Membership in our Society, one who has been a member of the Society for over thirty years, General William Wallace Atterbury."

Responding, President Durand spoke as follows:

"General William Wallace Atterbury: You have heard from the preceding speaker, reasons which have furnished the invitation for the present occasion.

"We have recognized in you a great engineer, a great organizer, and leader of men, a great director of human effort in the service of civilization; and for these reasons we have desired to honor you and in so doing to honor ourselves, by electing you to Honorary Membership in The American Society of Mechanical Engineers.

"This grade of Honorary Member is the highest and most honorable grade within our body, and is intended to be limited to great and outstanding leaders in our profession.

"With the greatest personal satisfaction and pleasure, therefore, General Atterbury, I give you official notice of this action on the part of The American Society of Mechanical Engineers, and I hand to you this diploma in certification of the same."

General Atterbury, in accepting the parchment evidencing the distinction that had been conferred on him, said:

"Mr. President, Fellow-Members, and Guests of The American Society of Mechanical Engineers:

"This is a great honor which you have just conferred upon me. It is one that fills me not only with a deep sense of gratitude to my friends and associates who have been so kind as to bring it about, but also with a most humble realization that Honorary Membership in this Society is a rare distinction.

"The certificate which our President has just handed me is held by less than a score of living men. Yet this organization is the largest engineering society in the world. Those who have been so honored in the past number a very small group. Therefore, that man would be bold, indeed, who did not enter such eminent company in a spirit of humility and appreciation.

"It is a happy circumstance that you are holding this meeting in Altoona and that the Pennsylvania Railroad is in a sense the host of the Society on this occasion. Many of you know, perhaps, that 39 years ago the Altoona shops of the Pennsylvania Railroad afforded me my first employment as a special apprentice. This community was my first railroad home. Consequently I feel that I may add to Mr. Lee's greeting a word of personal welcome.

"The American Society of Mechanical Engineers had been in existence then for about six years. Several years before, it had held one of its earliest meetings here. Now the Society is 45 years old. It has been my privilege to be a member of it for 31 years. During that period I have seen it become a live, active exponent of the ideals, the traditions, the obligations, and responsibilities of mechanical engineering as a leading profession in this country and throughout the world.

"Societies, like individuals, gain in experience with the years. It must be a source of real gratification to the members of this Society to realize that their collective experiences have accomplished outstanding results in the field of practical benefit to human welfare. The record of such achievement is long and noteworthy. It can perhaps best be summarized by saying that in letter and spirit this Society has lived up to that broad definition of engineering, which is 'the organizing and directing of men, forces and materials for the benefit of mankind.'

"The sessions upon which you have begun are to be devoted particularly to transportation. This is a field in which mechanical engineering has been particularly responsible for a great deal of the remarkable development that has occurred in railroading in the $4\frac{1}{2}$ decades of the Society's existence. It is enlightening even to those who have witnessed this development to realize that the science of transportation has undergone such rapid expansion and improvement in that comparatively short period. One of the anomalies of the day is that notwithstanding the perfection of modern transportation as compared with even a generation ago it is generally accepted as commonplace and regarded almost with indifference. Yet that fact alone is a tribute to the achievement which it signifies.

"Just as engineering science and skill in their practical application have wrought revolutionary changes in human society in general, so mechanical engineering in its own sphere has contributed much to the development of a system of railroad transportation in this country that is unrivaled the world over. In safety, in comfort, in convenience, in speed, in capacity and in actual service the science of railroad transportation has accompanied in its advance, the progress that has been made in engineering knowledge, methods, and procedure. It is not too much for us to say that this Society may properly measure its own achievements in terms of such outstanding accomplishments as I have indicated.

"The fact that this Society has existed for 45 years, advancing with constantly broadening purposes and ideals, with a membership of 18,000 in the United States and Canada, with 1500 members in active committee service, and with 13 professional divisions organized to cultivate the entire field of mechanical engineering—all this indicates its tremendous power and ability to serve the needs to which its members are devoted.

"However remarkable have been the strides of engineering progress in the past, the opportunities before us command all the experience, knowledge, breadth of vision, and foresight that this body represents. Many problems of practical importance have already become the subject of research projects under the Society's auspices. Railroad transportation alone invites your best ingenuity. We have only scratched the surface of improvements, economies, and efficiencies which almost every department of railroad transportation will require in the future.

"Fortunately, such tasks as these intrusted to the members of The American Society of Mechanical Engineers will command the attention of men of very high professional and intellectual attainments, and the results that we may expect cannot but be commensurate with what they and their associates in this Society have already accomplished.

"Mr. President, it is one of the great privileges of my life to accept Honorary Membership in The American Society of Mechanical Engineers. I want you and your associates in the Council of the Society, and the members at large, to know that I regard it as a high honor and that I am very grateful for this exceptional distinction."

Dr. Durand, in declaring the formal part of the meeting adjourned took occasion to say that the memory of the evening would live long with those present as a peculiarly interesting, significant and delightful occasion among the many memories which they would carry away from the regional meeting in Altoona.

The feature of the sessions was the address on Tuesday morning by Samuel Rea, Retired President of the Pennsylvania Railroad System, who took American Transportation as his topic. Mr. Rea's many contributions to transportation improvement during a lifetime of contact with transportation problems give him an unusual background from which to deal with this subject. His excellent address appears as the leading article in this issue of MECHANICAL ENGINEERING. Mr. Rea was followed by Lawford H. Fry, Metallurgical Engineer of the Standard Steel Works Company, who dealt with the Locomotive Testing Plant and Its Influence on Steam-Locomotive Design. Mr. Fry has acted as the chronicler of the achievements of the Altoona Test Plant since its installation in 1908. His authoritative and interesting paper appears in this issue of MECHANICAL ENGINEERING. The third speaker of the morning was Samuel P. Bush, President of the Buck-eye Steel Castings Company, of Columbus, Ohio, who spoke of the important contributions of the engineer to the transportation industry. Past-President Dexter S. Kimball presided at the Tuesday morning session.

At the Wednesday morning session, which was presided over by Roy V. Wright, member of Council and Vice-President elect of the A.S.M.E., three papers were given. A. J. County, Vice-President, Treasury, Accounting and Corporate Work, Pennsylvania Railroad System, traced the growth of the Pennsylvania Railroad System. Dr. M. E. McDonnell, Chief Chemist of the Pennsylvania Railroad System, in an important paper related recent progress in Rust-proofing of Materials. His paper appears in this issue. The final speaker was Robert S. Binkerd, Vice-Chairman of the Committee on Public Relations of the Eastern Railroads, who dealt with The Railway as an Economic Force. The papers by Mr. Bush, Mr. County and Mr. Binkerd will appear in the December issue of MECHANICAL ENGINEERING.

Book Reviews and Library Notes

THE Library is a coöperative activity of the A.S.C.E., the A.I.M.E., the A.S.M.E. and the A.I.E.E. It is administered by the United Engineering Society as a public reference library of engineering and the allied sciences. It contains 150,000 volumes and pamphlets and receives currently most of the important periodicals in its field. It is housed in the Engineering Societies Building, 29 West 39th St., New York, N. Y. In order to place its resources at the disposal of those unable to visit it in person, the Library is prepared to furnish lists of references to engineering subjects, copies of translations of articles, and similar assistance. Charges sufficient to cover the cost of this work are made.

The Library maintains a collection of modern technical books which may be rented by members residing in North America. A rental of five cents a day, plus transportation, is charged. In asking for information, letters should be made as definite as possible, so that the investigator may understand clearly what is desired.

The German Museum of Engineering, Science, and Industry

DAS DEUTSCHE MUSEUM. A memorial volume prepared under the direction of Dr. Conrad Matschoss. V. D. I. Verlag, Berlin, and R. Oldenbourg, Munich, 1925. Cloth, 8¼ × 11¼ in., 364 pp., profusely illustrated. 20 marks.

DEDICATION of the Deutsche Museum on May 17, 1925, on the 70th birthday of its founder, Dr. Oskar von Miller, brought to a climax 22 years of earnest work in an unselfish project for the upbuilding of German engineering, industry, and education. This volume is issued by the Verein Deutscher Ingenieure in honor of that epoch-making event. The material was prepared under the direction of Dr. Conrad Matschoss, engineering educator, discerning historian of engineering achievements, and friend of many American engineers.

The happy method has been adopted of bringing together a series of monographs which present the illuminating story of the founding of the project, the realization of the plan, and an adequate though necessarily condensed description of the museum exhibits. Throughout there is generous mention and credit for those who made the undertaking possible, and honor to the scientists and engineers whose discoveries have made engineering the master of the forces and resources of nature.

The book opens with a greeting to the museum from the Verein Deutscher Ingenieure, dated on the day of dedication.

The first monograph is by Dr. Matschoss, in which he sketches the growth of the museum idea, and the influence of the Verein in its development, and pays tribute to Dr. von Miller as the one whose foresight, energy, and leadership brought reality out of a great vision. The museum is and always will be a monument to his ability and tireless effort.

Dr. G. C. von Linde has enriched the account of the building of the museum with flashes of the personalities of many of those who supported the movement during the twenty-two years from June, 1903, to May, 1925. The raising of money and securing of donations was a continuing problem to which Dr. von Miller brought a genius for solution and accomplishment.

The first room of the museum opened to the reader is the Hall of Fame (Der Ehrensaal). In an early paragraph of its description Dr. Walther von Dyck quotes a line from Goethe: "Das beste Monument des Menschen ist der Mensch." In this formal hall are portraits and portrait busts of many whose memory the world honors for their great discoveries. To name but a few from the many, honor is thus bestowed upon: Henne Gensfleisch (Gutenberg), Kepler, Leibniz, Fraunhofer, Gauss, Ohm, Siemens, Bunsen, Helmholtz, Hertz, Borsig, Krupp, Lilienthal, and Zeppelin.

The controlling thought in the minds of those who have promoted the Deutsche Museum was to establish a living educational institution, not a mere depository for relics. This aspect is discussed by Dr. Georg Kerschensteiner in connection with the plan of arrangement of the museum, its pedagogical value, and methods of use.

To Dr. J. B. Bosch was intrusted the story of the museum building and building the museum. The site is an island in the river Isar in a central location in the city of Munich. The building is of reinforced concrete, with each section specially designed for the objects to be housed and displayed. The floor area is approximately 250,000 sq. ft.; the cost was about 15,000,000 gold marks ($3,700,000). This part of the book is fully illustrated with plans and details and is enriched with reproductions of some of the great mural paintings.

With the exception of the closing monograph, the rest of the book is given over to a description of the main divisions of the exhibits. Turning to the final pages first, it is found that with rare but characteristic modesty they have been taken by Dr. von Miller. With complete generosity he tells of those who supported the project in the early years, and of their assistance by contributions of money and donations of materials; of the securing of some of the more striking exhibits until there are now nearly 60,000 objects on view; and closes with a dedication of the great institution to all Germans of whatever section or party—unselfish to the end.

The subdivisions of the exhibits briefly described and shown by representative objects or groups are these:

Mining and Metallurgy	Astronomy
Prime Movers	Textile Manufacture
Roads and Railroads	Papermaking
Bridges	Graphic Arts
Shipbuilding and Harbor Works	Agriculture
Aircraft	Brewing and Distilling
Mathematics	Building Construction
Physics	Water Supply
Electricity and the Telegraph	Heating
Musical Instruments	Electrotechnics
Chemistry	Lighting

Each of these sections is enriched with priceless original models and first operating devices and machines of epoch-making discoveries and inventions—as examples the first Diesel engine of 1897 and the first Zoelly steam turbine of 1903. There are also numerous replicas of other great inventions made, with perfect reproduction of even the minutest details. Examples are a Watt engine of 1788 and "Puffing Billy," the locomotive of William Hedley built in 1813.

Other exhibits show the development of an art from the earliest days down to the present time. A striking example of this character is the various stages in the development of the plow. At one of this life-size group is a slave dragging a crooked stick to scratch a shallow furrow. Then a group of three slaves dragging a crude wooden plow, guided by a fourth. Then successively other wooden plows drawn by animals—buffaloes, oxen, and horses—until we come to modern farm implements used with animal power. Then another part of the exhibit takes up the development of power-driven farm machinery, modern plows, harrows, and tractors. Thus the visitor at a glance can see the principal stages of development in the art of constructing and using tools and machinery for plowing. This is without doubt one of the most helpful types of exhibit. Others worked out with equal thoroughness in the Deutsche Museum show the rise, progress, and present status of the great arts which form the foundation of modern industry.

No American reader can close this memorial volume without the deep hope and earnest wish that we may some day have a counterpart of this engineering and educational institution for the benefit of the youth of America.

L. P. ALFORD.[1]

[1] Editor, *Management in Manufacturing*, New York, N. Y. Mem. A.S.M.E.

A Directory of Specifications

NATIONAL DIRECTORY OF COMMODITY SPECIFICATIONS, 1925. Department of Commerce, Bureau of Standards, Miscellaneous Publication No. 65. Government Printing Office, Washington, D. C. Cloth, 7½ × 10½ in., 374 pp. $1.25.

THE National Directory of Commodity Specifications which has been under compilation for over two years by the Department of Commerce, has appeared as Miscellaneous Publication No. 65.

This volume represents the first attempt on the part of the Department of Commerce to collect and publish a classified list of existing commodity specifications promulgated not only by the different departments of the Federal Government, but also by the nationally recognized trade associations, technical societies, and leading public utilities.

According to Secretary Hoover in the Foreword, the ultimate aim of this publication is to bring about, through the wider use of reliable specifications, the unification of demand in large purchases, to the end of raising and stabilizing the quality of American production.

Making information regarding the extensive specification resources of the country available in this way will surely have most useful results in many directions—in bringing about a wider use of specifications, particularly among groups that are not now fully conversant with the great advantages to be gained by purchases on a specification basis; and secondarily, through the attention the long lists in this volume will call to the evidently unnecessary diversity that exists in specifications for the same commodity for identical or substantially identical use. Certainly a large waste of money and effort in purchasing, manufacture, inspection, and testing must result from this surprising elaboration of specifications, now for the first time really apparent.

There are listed almost 27,000 specifications for more than 6000 commodities, covering the following commodity groups: animals and animal products; vegetable food products other than oil seeds, expressed oils, and beverages; other vegetable products (except food, fibers, and wood); textiles; wood and paper; non-metallic minerals; ores, metals, and manufactures; machinery and chemicals and allied products; miscellaneous.

A statement of endorsement by the Advisory Board under whose supervision the preparation of the Directory went forward, prefaces the classified list of specifications.

The American Engineering Standards Committee has agreed to act as the central agent in the distribution of the technical-society and trade-association specifications listed, for those who wish to obtain them from one rather than from several sources.

The Directory is obtainable from the Superintendent of Documents, Government Printing Office, Washington, D. C. for $1.25 a copy.

Principles of Machine Design

PRINCIPLES OF MACHINE DESIGN. By C. A. Norman. Macmillan Company, New York, 1925. (Engineering Science Series.) Cloth, 6 × 9 in., 710 pp., illus., diagrams, tables, $6.50.

THE author of this book makes a particularly noteworthy effort to give the student and engineer engaged in the study or practice of modern machine design a greater incentive to "constructive thinking" by furnishing many illustrations of established or commercial designs. Soundness and uniformity in current practice are promoted to a limited extent by the author in drawing information freely from the standards and codes of the A.S.M.E., S.A.E., and other organizations. For instance, in Chapter III, on riveting and welding, the section of the A.S.M.E. Boiler Code on joint efficiency is given, although in Chapter V it is a disappointment not to find a similar recourse to the work done by the various A.S.M.E. standardization committees on screw threads and bolts.

In the introduction to Chapter I the author very ably states that in practicing machine design "we may concentrate our attention on the development of a new type of machine to perform a certain function; or we may find our main objective in the production of more or less standardized machinery at a minimum possible cost. From this angle, machine design is inextricably bound up with shop production and production methods." It might be pointed out here that in the development of a new type of machine there will be machine elements also more or less standardized. The author goes on further to state that although the designer may be limited, for commercial reasons, in laying out a machine of highest engineering excellence, the final product must be "fundamentally sound and valuable" if it is to have enduring success.

The remainder of Chapter I is devoted to the strength and failure of materials, and includes a brief exposition of the classical controversy between the maximum-shear and the maximum-strain theories. In view of the more practically established methods presented in the other chapters of the book, however, this discussion seems to be out of harmony. True as it is that the method of failure has an important bearing on machine design, it hardly seems wise to concern the practical designer with theoretical causes of a controversial nature in the very first chapter. Further, the impression that stiffness and strength mean one and the same thing should be corrected.

The second chapter, on materials, is a very important one, since it covers the principal light-weight materials and the strength of metals at elevated temperatures. The table giving the strength properties of about forty important engineering metals should prove to be of value.

In Chapter VIII, on cylinders and cylinder heads, the design of the Liberty-engine and the Puma-engine cylinders is discussed. The chapter dealing with the design of spur gearing treats of such important problems as gear cutting, the avoiding of tooth interference, sliding of gear teeth, and gear materials. No mention is made, however, of the enveloping form of teeth developed recently in England by Bickers, Bostock & Bramley, which is supposed to reduce or eliminate slip entirely. Lewis' and Barth's formulas and a table of wear factors for different materials and pitch-line velocities are given. Separate chapters are likewise devoted to helical, bevel, worm, and spiral gears.

On the whole, the book is a valuable treatise on the subject of machine design in which the various machine elements are covered by separate chapters, notable among which are those on bearings and brakes.

JOSEPH KAYE WOOD.[1]

Books Received in the Library

A.S.T.M. STANDARDS ADOPTED IN 1925. American Society for Testing Materials, Philadelphia, 1925. Paper, 6 × 9 in., 117 pp., $1.50.

This pamphlet, the first supplement to the 1924 Book of A.S.T.M. Standards, contains thirty-six standards adopted in August, 1925, and replacements for two methods that appeared in the Standards. These additions include, among others, specifications for copper, trolley wire, condenser tubes, gypsum and gypsum building materials, asphalts and pitches for water-proofing and for tire fabrics.

APROVECHAMIENTO DE LAS ENERGIAS NATURALES. By Juan Gelpi Blanco. Published by the author, Barcelona, 1924. Cloth, 7 × 10 in., 283 pp., diagrams, tables, 24 pesetas.

This treatise deals with the utilization of power resources, with special reference to conditions in Spain. It discusses the natural sources of power—winds, waves, tides, streams, and fuels—the importance of efficient utilization, the results achieved in modern hydroelectric plants, heat engines, and electric transmission of power. The power resources of Spain are inventoried and attention is called to the need for conservation. Throughout, the subject is treated practically, from the point of view of the engineer and the manufacturer.

BEITRÄGE ZUR GESCHICHTE DER TECHNIK UND INDUSTRIE; Jahrbuch des Vereines Deutscher Ingenieure. Vol. 14, 1924. Edited by Conrad Matschoss. Published by V. D. I., Berlin, 1924. Paper, 7 × 11 in., 278 pp., illus., diagrams, portraits, $1.75.

The sixteen historical papers in the 1924 volume cover a wide range. There are articles on the history of railroad brakes, of wine presses, of screw threads, and of chemistry. Biographical contributions include an account of Alfred Krupp as a machine designer; a life of B. H. Strousberg, a German "railway king;"

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an article on C. Waltjen, one of the pioneers of iron shipbuilding in Germany; and a description of the industrial activity of Emil Kessler, one of the first German locomotive builders. Other articles discuss the history of aluminum, the evolution of wire-drawing machinery, cranes, accumulators, and traps and spring guns. Like its predecessors, the volume is valuable to all who are interested in the history of industry.

COPPER AND ALLOYS. By Charles H. Hughes. Chart, 17 × 21½ in. Published by the author at 2681 Amboy Road, New Dorp, S. I., New York, N. Y. Cardboard, $1.25; paper, $1.00.

A chart showing in the first section general data as to copper as such, namely, the grades, commercial forms, general, physical, chemical, and electrical properties, and effects of different metals on copper alloys and of copper in different metals. The next section gives a list of the commercial copper alloys with the proportions of the alloying metals in percentages and data of physical properties, weights, melting points, etc. Tables of gages for sheet and tubes are also given.

DRAFTING METHODS. By Douglas S. Trowbridge. Codex Book Co., New York, 1925. Cloth, 5 × 8 in., 155 pp., illus., $2.50.

The scope of this small reference book is indicated by its contents. The suggestions given and the practices described are intended to promote good practice and efficiency in the drawing room. The book is not intended as a textbook but rather as a guide to convenient methods and rapid ways of working.

ECONOMICS OF PUBLIC UTILITIES. By L. R. Nash. McGraw-Hill Book Co., New York, 1925. Cloth, 6 × 9 in., 430 pp., diagrams, tables, $4.00.

The aim of the author has been to assemble the essential facts concerning the broader business and economic problems of our public utilities and to analyze and discuss these facts. The discussion is confined to the problems of electric light and power, electric-railway, and gas properties. It is intended for executives, engineers, city and state officials, and others who need a knowledge of the fundamentals of the industry.

EISENBAHNWESEN; Die Eisenbahntechnische Tagung und Ihre Ausstellungen, 1924. Edited by Conrad Matschoss. Published by V. D. I., Berlin, 1925. (Sonderausgabe der *Zeitschrift des Vereines deutscher Ingenieure.*) Cloth, 9 × 12 in., 614 pp., illus., diagrams, plates, 9 × 12 in., cloth.

In 1924 the Verein Deutscher Ingenieure adopted a plan for meetings devoted to special branches of engineering, in addition to the general annual meetings. One of these, on railroad engineering, occurred at Berlin, September 21–27, 1924, which was attended by over 5000 railroad men. The papers presented there and the discussions that they provoked are published in this large, bound special number of the *Zeitschrift* and form a good survey of the present state of railroad engineering in Germany.

Among the topics discussed are methods for improving the thermal economy of locomotives, turbine locomotives, the design of electric locomotives, the thermo-locomotive, transportation of bulk freight in dumping cars, electric-railroad operation, brakes, ball and roller bearings, powdered-coal firing, chilled wheels, American signal systems, classifying yards, standards, and modern bridges and tunnels. There is also a description of the exhibition of railroad equipment held in connection with the meeting and a good bibliography of the German books on railroad topics published during the past decade.

ELEMENTARY MATHEMATICAL ANALYSIS. By Charles S. Slichter; reset with revisions and additions by Warren Weaver. Third edition McGraw-Hill Book Co., New York, 1925. (Modern mathematical texts.) Cloth, 5 × 8 in., 473 pp., $3.

This is not intended to be a text on "practical mathematics" in the sense of making use of scientific material and of fundamental notions not already in the possession of the student or in the sense of making the principles of mathematics secondary to its technique. On the contrary, the aim is to give the fundamental truths of elementary analysis as much prominence as seems possible in a course for freshmen. The emphasis of the book is placed upon the notion of functionality. Illustrations from science are freely used to make this concept prominent, and the student is taught to use mathematics to express and interpret the laws of actual phenomena, not merely to secure here and there certain computed results.

ELEMENTARY MECHANICAL DRAWING. By Charles William Weick; revised by Frank C. Panuska. Second edition. McGraw-Hill Book Co., New York, 1925. Cloth, 6 × 9 in., 251 pp., illus., diagrams, $2.

This textbook has been revised in order that the text and problems might conform to the recent developments in the teaching of the subject. A new chapter on sketching has been added and important changes made in various parts of the text. The book is intended to cover theory and practice and to proceed regularly from the elements through the fundamental training needed for general practice in the drafting office.

GRAPHIC TABLE COMBINING LOGARITHMS AND ANTI-LOGARITHMS. By Adrien Lacroix and Charles L. Ragot. Macmillan Co., New York, 1925. Cloth, 7 × 10 in., 56 pp., $1.40.

The usual tables of logarithms have the disadvantage that they are essentially "one way" tables, hence interpolation is necessary to find the numbers corresponding to the results obtained in computations. They also give the five-place logarithms of only those numbers that have four places. The five-place table in this book combines all five-place numbers and all five-place logarithms in one graphic scale from which either can be read in terms of the other, without interpolation. This arrangement makes the table more convenient and reliable than the customary form and enables it to be used more rapidly, the authors state. It also saves, they say, eighty per cent of the space.

HYDRAULICS AND ITS APPLICATIONS. By A. H. Gibson. Third edition. D. Van Nostrand Co., New York, 1925. Cloth, 6 × 9 in., 801 pp., illus., diagrams, tables, $6.

While primarily a theoretical work on hydraulics, Dr. Gibson's book is intended to consider the application of that theory to the design of hydraulic machinery, in a manner suitable for a student with some knowledge of mechanics. The book opens with prefatory chapters on the physical properties of water and on hydrostatics. The second section, hydraulics, has chapters on the flow of fluids, the motion of fluids in pipes, bends, orifices and vortices, skin friction, the impact of jets, pressure on submerged planes, resistance of ships, waves and ripples. In the third section hydraulic machinery is taken up. The various forms of water wheels and turbines are described and the theory of turbine design is discussed. Other chapters treat of the hydraulic engine, pumps, rams, air compressors, hydraulic power transmission, meters, cranes, presses, riveters, jacks, brakes, dock machinery, etc.

ICARUS, or The Future of Science. By Bertrand Russell. E. P. Dutton & Co., New York, 1924. (To-day and to-morrow series.) Cloth, 5 × 7 in., 64 pp., $1.

Mr. Russell thinks that science threatens to cause the destruction of our civilization. This little book, which is to a certain extent a reply to Daedalus, sets forth some of the dangers inherent in science while we retain our present political and economic institutions.

MANUAL OF ORGANON, THE PHILOSOPHY AND SCIENCE OF ORGANING THE ART OF MANAGEMENT. By Charles W. Gremple. Organon Lyceum, New York, 1925. Fabrikoid, 32 pp., 4 × 7 in., $2.

These pages, according to the preface, "form a condensation of considerable material collected and developed by the author an investigation and experimentation carried on during the past fifteen years." In the view of the author, formal management should not be treated as a branch of engineering but as an independent art, the art of utilizing disciplined procedure in any human endeavor. The present work is concerned with the discovery of the basic laws underlying management, not with the various possible methods for applying them.

MARINE STEAM ENGINE. By Richard Sennett and Sir Henry J. Oram. Fourteenth edition. Longmans, Green & Co., London and New York, 1924. Cloth, 6 × 9 in., 478 pp., illus., diagrams, $8.50.

The popularity of this well-known text is indicated by its forty years of life, during which time fourteen editions have appeared. The last previous revision was in 1916. The present edition follows the plan of earlier ones and apparently there has been little

change in most of the text, but a new chapter on the geared steam-turbine has been added, and a chapter on theoretical indicator diagrams, which appeared in the thirteenth edition, has been omitted.

PHYSICAL METALLOGRAPHY. By E. Heyn; translated from the German by Marcus A. Grossmann. John Wiley & Sons, New York, 1925. Cloth, 6 × 9 in., 440 pp., illus., diagrams, tables, $6.

This work makes readily accessible to American metallurgists the investigations of Professor Heyn and his conclusions. Throughout the book emphasis is laid on the multiplicity of sciences involved in metallography. After a general survey of metals and alloys, the book deals with the developments during the freezing and cooling of alloys, the determination of equilibrium diagrams, the structure of metals and alloys and the methods of observing it, properties of hardness and strength, metallic substances and gases, shrinkage and attendant phenomena, magnetic properties, and electric conductivity. The translator has added a chapter on steel and has also called attention to recent advances and developments. A bibliography accompanies each chapter.

PRACTICAL STEAM, HOT WATER AND VAPOR HEATING AND VENTILATION By Alfred G. King. Norman W. Henley Pub. Co., New York, 1925. Cloth, 6 × 9 in., 551 pp., illus., tables, $4.

A practical book on heating and ventilation, which gives an account of the principles involved, the apparatus used, and the methods of installation, in a form suited to the needs of plumbers and steam-fitters engaged in estimating and installing systems.

PRINCIPLES OF INDUSTRIAL ORGANIZATION. By Dexter S. Kimball. Third edition. McGraw-Hill Book Co., New York, 1925. Cloth, 6 × 9 in., 436 pp., illus., $4.

Intended as a textbook on the science of management, this book is devoted to a discussion of general principles and does not attempt to illustrate the many kinds of forms used in industrial management. The volume opens with an account of the origin and growth of the important features of industrial organization. The author then discusses the important general questions, such as the location and arrangement of plants, the principles of organization, coördination and executive control, the control of production, the standardization of operation, the compensation of labor, purchasing, inspection, cost finding, and personnel administration.

PRINCIPLES OF LOCOMOTIVE OPERATION. By Arthur Julius Wood. Second edition. McGraw-Hill Book Co., New York, 1925. Cloth, 6 × 9 in., 315 pp., illus., diagrams, tables, $3.

In the ten years that have passed since this textbook on locomotive performance first appeared, many new developments have occurred, and the author has taken advantage of this revision to add a number of new topics. Among them are new constants for calculating the tractive force at any speed, a method for determining the maximum speed for rated tractive force, problems in foundation, brake rigging, time lost in slowdowns, the booster, a method for laying out a Walschaerts valve gear; factors and ratios for empirical design, calculations for counterbalancing, and the modernizing of the locomotive.

DIE STATIK DER BRÜCKENKRANE. By J. M. Bernhard. R. Oldenbourg, Munich and Berlin, 1925. Paper, 7 × 10 in., 51 pp., plates, diagrams, 3.80 marks.

This pamphlet is intended to help designers of bridge cranes by illustrating the newest methods for their calculation. The author has selected two common types and has worked out in full the static and analytic calculations involved. The book is a convenient compilation of the formulas and other data required by the designer.

TESTS FOR RAILWAY MATERIAL AND EQUIPMENT. By Henry Knauer. Simmons-Boardman Publishing Co., New York, 1925. (Railwaymen's handbook series.) Fabrikoid, 5 × 8 in., 257 pp., illus., diagrams, $3.

An elementary book, intended for those taking up work in the laboratory and for those in the shops who wish to inspect railroad materials in the field. The materials discussed include water, paint and varnishes, oils, non-ferrous metals, rubber goods, cement, iron, steel, and fuel. Tests of boilers and locomotives are also described. The phraseology is simple and does not require a knowledge of chemistry or higher mathematics.

TOUS LES ALLIAGES. By A. Chaplet. Gauthier-Villars et Cie., Paris, 1925. Paper, 6 × 10 in., 70 pp., 18 fr.

A convenient tabulation of some 2500 alloys, grouped by the principal metal, or, in some cases, by their use. The composition is given in each case. An index enables one to find an individual alloy.

UEBER WIRTSCHAFTLICHEN BRENNSTOFFVERBRAUCH IM EISENBAHNBETRIEBE. By H. Bager. Translated from the Swedish by W. Hausmann. Wilhelm Ernst & Son, Berlin, 1925. Paper, 7 × 10 in., 67 pp.

This brochure contains reports on the actual conditions which influence the coal consumption of locomotives, with some suggestions that may be of practical value in this matter.

The arrangements and methods which favor fuel economy are well known, but less is known of the economy or waste in specific cases. It is therefore one of the objects of this book to show, by numerical data from practice, how important fuel economy is in locomotive operation.

WÄRMESTROM-BILDER AUS DEM EISENHÜTTENWESEN. By Verein Deutscher Eisenhüttenleute. Verlag Stahleisen m.b.H., Düsseldorf, 1922. Paper, 8 × 11 in., 20 pp., $0.75.

The thirteen Sankey diagrams given in this pamphlet show graphically the flow of heat in various branches of iron works. From them the metallurgist learns at a glance how the energy consumed in any operation is distributed and what proportion of it is actually utilized. These diagrams are based on data collected by the Düsseldorf Wärmestelle during investigations covering three years. They cover such major topics as the thermal efficiency of coke ovens, old and modern blast furnaces, open-hearth furnaces, central producer-gas plants, forges, rolling mills, and wire mills.

WHAT THE COAL COMMISSION FOUND. Edited by Edward E. Hunt, F. G. Tryon, and Joseph H. Willits. Williams & Wilkins Co., Baltimore, 1925. Human relations series. 416 pp., cloth, 6 × 9 in., illus., tables, $5.

The work of the United States Coal Commission during 1922 and 1923 is of interest not only because of the recommendations but also because it represents one of the most exhaustive attempts to ascertain the facts concerning a great industry ever undertaken by our Government. The reports of the Commission cover a wide range of subjects and will fill several volumes when published. In the present volume, these reports are summarized briefly by certain members of the staff of the Commission and published in convenient form. The introductory section discusses the coal problem, bituminous and anthracite, and the need for a national fuel and power policy. In the second section are discussed the mining and selling of bituminous coal; costs, prices and profits; how the miners live, their wages, hours and earnings; the causes of strikes; and better methods of mining and transportation. The third section deals with anthracite mining along similar lines. Section four presents the recommendations of the Commission.

WHO'S WHO IN ENGINEERING. Second edition. By John William Leonard. Who's Who Publications, Inc., New York, 1925. Cloth, 6 × 10 in., 2483 pp., $10.

The second edition shows marked improvement over the first, particularly in size. Nearly one thousand pages have been added, and the book now contains brief biographies of more than eighteen thousand members of the engineering profession. The majority of these men are residents of North America, but there are also a considerable number from other countries. In addition to the alphabetical arrangement, a geographical index is supplied. The preface states that a conscientious effort has been made to include only names that are representative of the best in engineering.

Erratum

IN THE October, 1925, issue of MECHANICAL ENGINEERING on page 869 it was stated in the book note on An Introduction to Fluid Motion that the price was five dollars. The price of this book is one dollar and sixty-five cents.